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PROPERTIES OF TUNED VIBRATION DAMPERS

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UNIVERSITY OF MICHIGAN

OCTOBER 1952

Statement A
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WADC TECHNICAL REPORT 52-328

PROPERTIES OF TUNED VIBRATION DAMPERS

R. N. Hamme
University of Michigan

October 1952

Aircraft Laboratory
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Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base, Ohio

FOREWORD

This project was authorized by Research and Development Order No. 455-49, Aircraft Sound Control, and was accomplished by Contract AF18(600)-56 with the University of Michigan. The work was initiated by the Aircraft Laboratory, Directorate of Laboratories, Wright Air Development Center, Wright-Patterson Air Force Base, Ohio. Dr. O. R. Rogers of the Dynamics Branch, Aircraft Laboratory, initiated and monitored this project.

The members of the staff of the Acoustics Laboratory of the Engineering Research Institute, University of Michigan, who participated in the work on this project were: P. H. Geiger, who supervised the project, R. N. Hamme, H. F. Reiher, and H. T. Walsh.

ABSTRACT

Measurements are described which indicate the damping effectiveness of a light-weight vibration damping device which can be tuned to provide optimum damping over a selected frequency range. The criteria for comparison of spot dampers by means of vibration decay rate measurements are developed, and the relative damping effectivenesses of two types of spot damper are evaluated in terms of Air Force requirements. It is concluded that extraordinarily effective vibration dampers of this type can be fabricated for aircraft application pending a materials survey to determine the optimum compromise between damping effectiveness and other physical requirements.

PUBLICATION REVIEW

This report has been reviewed and is approved.

for R. G. RUEGG
Colonel, USAF
Chief, Aircraft Laboratory
Directorate of Laboratories

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PROPERTIES OF TUNED VIBRATION DAMPERS

For a number of years excellent vibration damping treatments have been available which will effectively suppress resonant vibrations of the panels on which they are applied. For the most part, these damping treatments function by dissipating a constant amount of vibratory energy during each cycle of vibration, the dissipation being due to internal friction during flexure of the damping material. Consequently, in order for the treatment to be effective at low frequencies, the dissipation per cycle must be increased either by increasing the flexure per cycle for a given damping material or by using damping materials with greater inherent damping. With the materials currently available, high damping can only be obtained at low frequencies by loading the damping material with mass elements, the inertia of which increases the flexure per cycle of vibration and therefore the damping. This technique cannot be used in damping aircraft fuselage panels, however, because effective damping at 50-150 cps (the frequency range over which panel resonances are most likely to be excited) would require too heavy a damping treatment. Furthermore, it would not be possible to avoid the weight requirement by the use of tuned vibration absorbers* because such devices provide protection at only a single frequency and suffer from the added disadvantage of causing vibration amplification at two neighboring frequencies.

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*The theory and application of tuned vibration absorbers have been extensively investigated and are reported at length in the literature. (See, for example, Den Hartog, Mechanical Vibrations, (McGraw-Hill, 1947) pp. 112-130) These devices depend for their effectiveness upon the generation of a large-amplitude vibration which is out of phase with the single vibration being suppressed.

It is therefore the purpose of this report to describe the initial exploratory work that has been done on a tuned vibration damper which gives promise of being suitable for aircraft application. The type of system under investigation is shown schematically in Figure 1. It consists of a "diaphragm" made of some material which has high internal damping, an "attached mass" which is fastened to the diaphragm, and a "spacer element" which provides clearance between the loaded diaphragm and the vibrating panel. When attached at an anti-nodal position to a vibrating panel, the reciprocating motion of the panel is transmitted through the spacer element to the outer edges of the diaphragm and the inertia of the attached mass causes flexure of the diaphragm. The relative magnitudes of the attached mass and the flexural stiffness of the diaphragm cause the amount of bending to change with frequency of vibration, hence the amount of energy dissipated per cycle is a function of frequency. The experiments described below provide a measure of the damping effectiveness of such a "spot damper" for various diaphragm materials and attached masses as a function of frequency of vibration.

Measurement of Damping Effectiveness

The apparatus used for measurement of the damping effectiveness of the spot dampers is shown schematically in Figure 2. The damper under test is attached to a steel plate which has dimensions selected to provide a fundamental natural frequency of vibration of the desired value. With the test plate selected for very low inherent damping and supported at nodal lines on very soft springs, the rate of decay of free vibration of the test plate with damper attached provides a relative measure of the damping effectiveness of the damper at the natural frequency of the test plate. The test plate is excited to resonant vibration by means of an electromagnet driven through a power amplifier by an oscillator which is tuned to the fundamental natural frequency of the test plate. The amplitude of excitation is immaterial over a wide range because the decay of free vibrations is logarithmic and the vibration decay rate is therefore independent of level. With the excitation turned off and the plate vibrating freely, a measurement of the rate of decay of sound pressure in decibels per second at a fixed point near the plate constitutes a measurement of vibration decay rate in db/sec. The amplitude of the decaying microphone signal is displayed as a horizontal line on a CR oscilloscope, and the logarithmic decay of this signal is photographed with a moving film camera. From the known spacing of amplitude marks on the oscilloscope face and the known film speed of the camera, the vibration decay rate can be computed in db/sec.

It is not enough, however, to investigate the damping effectiveness of a spot damper at a single frequency determined by the fundamental frequency of a single test plate. Because nothing is known of the dynamic behavior of spot dampers, it is not possible to deduce its damping effectiveness over any specified frequency range from measurements made at a single frequency, even if its behavior at this single frequency is investigated as a function of attached mass. The complete experimental analysis therefore requires a measurement of the damping effectiveness of the damper as a function of attached mass at several test frequencies.

In the early stages of the experimental work it was hoped that the fundamental natural frequency of a single test plate could be varied over the required frequency range by simple mass-loading of the testplate, the added mass being attached at the center of the plate in such a way that the attachment would not introduce appreciable damping. Experiments, however, soon revealed that such a procedure would be completely unfeasible for several reasons. The added mass, which was intended to lower the fundamental natural frequency of the plate, was found to suppress the amplitude of the fundamental so that excitation of the system became difficult without exciting higher harmonics. Furthermore, changing the added mass was found to cause an unpredictable migration of the nodal lines of the fundamental vibration so that nodal mounting of the test plate became difficult and dynamic equivalence of the vibrating system at the various test frequencies could not longer be assumed.

It is necessary, of course, in testing a damping treatment such as a spot damper which only partially covers the test plate, to preserve a dynamic equivalence among the various test frequencies; i.e., it is necessary to assure that a given spot damper has an equal chance to demonstrate its damping effectiveness at each test frequency by participating equally in the vibration of the test system at each frequency. It is obvious, for example, that a spot damper would be completely ineffective if it were attached to a vibrating system at a nodal position where it would not participate in the vibration of the system.

In order to test spot dampers at various frequencies without sacrificing the required dynamic equivalence, a series of test plates were selected, the dimensions of which were chosen to provide a series of fundamental natural frequencies which would cover the frequency range 50-200 cps in approximately equal steps. Since each of the plates was square and supported similarly, it was expected that the vibration patterns at the respective fundamental natural frequencies would be geometrically similar. Vibration surveys of each of the test plates with a nodagraph proved that this was substantially true: when vibrating freely at its fundamental natural frequency, each of the test plates had a nodal pattern geometrically similar to that shown in Figure 3. In each case, the central area enclosed by the nodal lines constituted closely the same percentage of the total plate area; hence, the envelope of the vibration amplitude plotted along any line across one of the test plates would be geometrically similar to that across any other, and the required dynamic equivalence is preserved from one test frequency to another.

The stringency of the above requirement is well demonstrated by a consideration of the significance of a measurement of vibration decay rate as a measure of damping. It can be easily shown that the vibration decay rate measured with a given system is related directly to the percentage of critical damping for the given system which the damping element contributes. The relation, derived under certain idealizing restrictions, is simply

$$\text{Decay Rate (Db/sec)} = 0.0868\,\omega_0 K$$

where ω_0 is the undamped natural frequency of the system* and K is the percentage of critical damping comprised by total damping present in the system. Inasmuch as comparing the percentage of critical damping furnished by different damping treatments when applied similarly to the same vibrating system is regarded as a valid comparison of the practical damping effectiveness of the various damping treatments, the comparison of vibration decay rates will also be a valid criterion if, for example, the conditions of the above proportionality are observed in the testing.

It is obvious that the test system should never be over damped during testing; hence it is necessary to select test

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* ω_0 is actually the circular frequency of undamped vibration; it is related to the true frequency f_o by $\omega_0 = 2\pi f_o$.

plates which are massive enough to prevent overdamping by the most effective damper which is to be tested. Furthermore, if damping effectiveness is to be measured at different frequencies, the test plates used at the various test frequencies must be of equivalent massiveness so that equally effective dampers will furnish the same percentage of critical damping when attached to the various test plates. Actually the critical damping of any system which can be idealized to a one-degree-of-freedom system depends upon the product of the equivalent lumped stiffness and the equivalent lumped mass of the system, while the natural frequency depends upon the ratio of these quantities. Consequently, if it is desired to obtain a series of test plates of different natural frequency, it is generally necessary to make a correction for differences in their massiveness before comparing decay rates measured with the different plates. With the dynamic equivalence demonstrated above, it is possible to evolve a very simple correction which depends only upon the total masses of the respective test plates used. This correction is deduced in Appendix I.

The specifications of the test plates selected to obtain the necessary frequency coverage are shown in Table 1. The respective colums are, reading from left to right: the fundamental natural frequency in cps, the edge length in inches, the thickness in inches, the vibration decay rate at the fundamental frequency in db/sec without a damper attached, and the ratio of the weight to the weight of the 180-cps plate (as used in the correction deduced in Appendix I).

Table 1

Schedule of Test Plates Used

Frequency	Edge Length	Thickness	Bare Decay Rate	Mass Ratio
180	20	0.301	0.85	1.00
146	20	0.257	0.93	0.85
129	24	0.345	0.92	1.64
113	23	0.252	0.65	1.10
84	18	0.109	1.31	0.29
67	25.5	0.192	0.55	1.03
51	30	0.191	0.39	1.43

The effects of the slight amount of damping inherent in the plates and their spring suspensions (represented by 1.31 db/sec at worst) were eliminated from the reported data by subtracting the appropriate bare-plate decay rate from the measured decay rate.

Preliminary tests

Inasmuch as no previous work has been done on the behavior of spot dampers, preliminary tests had to be conducted to determine the sizes and shapes of diaphragms that would be most expedient for extensive tests. In view of the aircraft requirement that the materials used be resistant to fungus, vermin, and fire and have damping characteristics that are relatively unimpaired by wide variations in temperature and humidity, a series of fibrous glass materials were obtained for test.

First, a circular spot damper, 4 inches in diameter, was constructed with both diaphragm and spacer ring made of one-fourth inch thick board which was cut by hand from 1 inch thick Fiberglas board. The diaphragm was cemented to the spacer ring with rubber cement and the spacer ring in turn was fastened to the center of a test plate with rubber cement. The variable attached mass consisted of a machine screw through the center of the diaphragm pulled up against thin sheet metal washers to hold snugly against but not to crush the Fiberglas; the mass is increased by adding washers. By varying the attached mass from zero to 25 grams, it was found to be impossible to tune this damper to any of the test plates; i.e., no optimum attached mass could be found for which the vibration decay rate on any plate was higher than that measured with greater mass and less mass, respectively. This behavior was attributed to lack of stiffness in the diaphragm, the diaphragm material being so weakened in cutting it from thicker material.

A 3 inch diameter circular spot damper constructed of the same materials could, however, be tuned to the 146-cps plate, but the behavior of this damper could not be investigated as a function of frequency because a change of frequency requires the removal of the damper from the test plate. This operation of removal and reattachment invariably either ruined the damper or damaged it so severely that its damping properties would undoubtedly change from one test plate to another.

By this stage of the experiments other fibrous glass materials became available; namely, a series of "high-density" fibrous glass materials manufactured by the Vibradamp Corporation. The materials ranged in density from 10 to 20 lbs/ft^3 and in thickness from 1/16 to 1/4 inch. Another series of "high-density" Aerocor "B" Fiberglas samples were obtained ranging in density from 2.4 to 19.2 lbs/ft^3 and in thickness from 1/16 to 1 inch. Since the samples were furnished in 6 inch squares, several of the materials were tested as spot-damper diaphragms of this size, the spacer element consisting of two 1/4" x 3/8" x 6" wooden strips attached at opposite sides of the square diaphragms with machine screws through sheet metal retaining strips. The wooden strips were attached to the respective test plates with strips of "Scotch" brand Pressure Sensitive Tape No. 400 Double-Coated Tissue made by the Minnesota Mining and Manufacturing Company (hereinafter referred to as double-coated Scotch tape). Using 6 inch square diaphragms, it was not possible, using any of the above materials, to tune the spot damper at any of the test frequencies with attached masses not exceeding 25 grams.

However, it was found possible to tune 3 inch diameter circular spot dampers cut from several of these materials to frequencies within the range of investigation with attached masses of reasonable weight. Of the several dampers tried, a 3 inch diameter circular spot damper made of Vibradamp 10 lb/ft^3 was selected for extended testing. This selection was made solely on the basis of two considerations: (1) the stiffness of a 3 inch diameter diaphragm of this material permitted tuning of the spot damper to a clearly defined optimum damping effectiveness at the highest test frequency with only 15 grams attached mass, the (2) the surface of this material was smooth and rugged enough to stand the removal and reattachment of Scotch tape several times without great damage to the material.

Discussion of Experimental Results

The experimental data obtained with the 3 inch diameter spot damper made of Vibradamp 10 lb/ft^3 are shown in Figures 4-7 where the vibration decay rate in db/sec relative to the 180-cps testplate is plotted against attached mass in grams for 180, 146, 129, and 113 cps respectively. The damper consists of a 3 inch diameter Vibradamp diaphragm 1/4 inch thick and a 3 inch outside diameter 2 inch inside diameter Vibradamp spacer ring 1/4 inch thick. The diaphragm and spacer ring are rubber cemented together permanently and the unit is attached to the respective test plates with four tabs of double-coated Scotch tape. The variation of attached mass is accomplished by attaching various combinations of brass washers to a 5-40 x 1" machine screw which passes through the center of the diaphragm. The machine screw is pulled up and locked against two sheet-tin washers 1/2 inch in diameter so that the attached masses are firmly retained but the Vibradamp material at the center of the diaphragm is not crushed. In all but the first test, a standard series of attached masses was used at each frequency, the schedule being coverage in 2 gram steps with combinations of five standard masses of nominal weight 2, 4, 4, 10 and 20 grams followed by coverage in 0.2 gram steps where further detail seems indicated.

The curves of Figures 4-7, which are summarized in Figure 8, show extremely sharp tuning at each of the four frequencies investigated. As to be expected, greater attached masses are required to tune the damper to lower frequencies. Though the data of Figure 8 demonstrate the qualitative behavior of spot dampers well enough, the survey was not continued to lower frequencies for several reasons. First, this particular spot damper tunes too sharply for practical application: at 180 cps the vibration decay rate changes from 10 db/sec to a peak of 170 db/sec and back down again to 10 db/sec as the attached mass is changed from 11 grams to 25 grams. Consequently, if this damper is tuned for optimum damping at a given frequency it provides very little damping at other frequencies; e.g., if the damper were tuned to provide optimum damping at 113 cps by attaching 35 grams, it would provide only 30 db/sec at 129 cps and less than 10 db/sec at 146 cps and above. Secondly, the material of this damper is too delicate to permit reproducibility of attachment from one test plate to another so that the experimental data cannot be obtained which would be necessary to plot damping effectiveness versus frequency and, more particularly, to plot frequency of optimum damping effectiveness against attached mass, the curve from which any tuning law must be deduced. Difficulty was even experienced in changing the attached mass without permanently changing the inherent damping properties of the damper; this difficulty is reflected in the extrapolated portion of the 113 cps curve for this damper.

These objections to dampers made of fibrous glass materials were resolved by going to asphalted felt in the construction of further spot dampers, despite the poorer temperature stability to be expected of asphalted materials. The data obtained on a 3 inch diameter circular spot damper with both diaphragm and spacer ring made of a 2 ply indented asphalted felt manufactured by the Ruberoid Corporation are shown in Figures 9 and 10 for the two frequencies 180 and 146 cps respectively. No measurements were made at other frequencies because it is apparent that optimum damping would require attached masses exceeding 45 grams at all lower frequencies. In Figure 11 the behavior of the asphalted felt damper is compared with that of the Vibradamp damper. Both tests were made at 180 cps with dampers of the same diameter. It is evident that the asphalted felt damper is nearly as effective as the Vibradamp damper of the same size at its optimum loading, and in addition it is much less sharply tuned so that effective damping will still be provided at frequencies considerably above the tuned frequency in contrast to the behavior of fibrous glass dampers. This advantage is demonstrated more clearly as a function of frequency in the extended tests which follow.

It being desired to obtain an asphalted felt damper which would tune to 180 cps with much smaller attached mass so that the migration of the tuning peak could be demonstrated over a wide frequency range, the diameter of the spot damper was increased to reduce diaphragm stiffness. The result is shown in Figure 12 where otherwise identical asphalted felt spot dampers of different diameter are compared at 180 cps. An increase in diameter from 3 to 3-1/2 inches has reduced the attached mass required for tuning at this frequency from about 28 grams to 15 grams. It has also reduced the damping effectiveness at the peak by a considerable amount.

The complete series of tests on the 3-1/2 inch diameter asphalted felt damper is shown in Figure 13-17 and is summarized in Figure 18. As frequency is decreased, more and more attached mass must be added to tune the damper to optimum effectiveness, but there seems to be little trend for peak damping effectiveness to fall off at lower frequencies. The effectiveness of this damper is interpreted directly in terms of frequency in Figure 19, though the five frequency points available allow a great deal of freedom in drawing the damping versus frequency curves. In this figure, the damping effectiveness of the damper is plotted against frequency for several different attached masses. For example, if 25 grams is attached, little damping is provided below 94 cps (the 10 db/sec value), but damping increases rapidly above this frequency to a peak value of about 60 db/sec at 125 cps. The fall-off in damping effectiveness above the peak value is very gradual, however, so that adequate damping is still provided at the higher frequencies. To increase the effectiveness of this damper at the lower frequencies, it is necessary to increase the attached mass and, unfortunately, a situation of diminishing returns seems to be developing in this respect as shown in Figure 19. The amount by which the frequency of peak damping is shifted toward lower frequencies becomes smaller per equal increase of attached mass as the total attached mass increases.

The migration of peak frequency is plotted directly in terms of attached mass in Figure 20. Only three points are available, so very little more than the requirement of increased mass for tuning to lower frequencies can be demonstrated. The attached masses required to produce two arbitrarily selected amounts of damping are also plotted against frequency in Figure 20; e.g. if 10 db/sec vibration decay rate is required at a given frequency (say 120 cps) then the minimum attached mass that can be used with this particular damper can be read off the ordinate (15 grams for 120 cps). The data of Figure 18 then guarantees that greater than this amount of damping will be provided at all higher frequencies up to a least 180 cps and probably well beyond.

When the data of Figure 20 are plotted on log-log paper in an effort to determine an analytic tuning law, the spread of data from a straight line is greater than the measurement errors. This was attributed to permanent changes being made in the damper in changing it from one test plate to another so that damping-versus-frequency data here is not reliably indicative of a damper once attached and left undisturbed. Since this type of damage occured during removal due to flexing the diaphragm well beyond its elastic limits, it was decided to replace the flexible asphalted felt spacer ring by a metallic ring which would not bend during removal from the testplate. The effect of this change is shown in Figure 21 where otherwise identical spot dampers are compared with different spacer-ring materials. Since the only function of the spacer ring is to transmit vibrational energy to the damping diaphragm, it is not surprising that the rigid ring accomplishes this function somewhat better than the flexible ring without appreciably affecting the frequency of tuning which is connected only with diaphragm stiffness.

The data obtained on a 3-1/2 inch diameter circular spot damper with asphalted felt diaphragm and aluminum spacer ring are shown in Figures 22-27 for the respective test frequencies, and the data are summarized in Figure 28. None of the qualitative aspects of the family of curves has been changed by the use of a rigid spacer ring. When the damping effectiveness is plotted against frequency in Figure 29, however, there seems to be somewhat less of a diminishing return in lowering the tuning frequency by increasing the attached mass and a somewhat increased tendency to maintain effective damping at higher frequencies for given attached mass.

Though only four points are available for this damper in plotting tuning frequency against attached mass, one surprising result follows from a consideration of the graphs of attached mass versus frequency for given damping effectiveness as plotted in Figure 30. In this figure, as in Figure 20, the minimum attached mass that can be used to obtain a given amount of damping at a given frequency is plotted for two arbitrarily chosen values of damping effectiveness in addition to peak effectiveness. If the data of this figure are replotted on log-log paper, Figure 31 is the result, where the spread of points about the straight line labeled 10 db/sec is within experimental error. Lines drawn parallel to the 10 db/sec line (which is determined by the most reliable set of points) provide a close fit to the points measured at 30 db/sec and peak damping. The slope of these lines is -1.84 implying a tuning law:

$$mf^{1.84} = k \qquad \text{or} \qquad f = \sqrt[1.84]{\frac{k}{m}}$$

where k is a constant related to the damping effectiveness through the stiffness of the diaphragm. Hence, insofar as this procedure is valid in determining a tuning law, the frequency of optimum damping is predicted closely by assuming it to be inversely proportional to the square root of the attached mass. There is no indication at present whether this simple approximation is valid outside the frequency range investigated, for other shapes of damper, or for other materials. Accurate measurements are indispensable in such a determination, and the accuracy and reproducibility of the present results are sharply limited by the necessity of detaching the damper from its test plate for every change of frequency.

The changes in the effectiveness of the two types of damper tested when the ambient temperature is varied over a wide range are shown in Figures 32 and 33. These measurements were made at a single frequency (180 cps) by tuning the damper in question to near-optimum effectiveness at room temperature by attaching the proper mass and then measuring the damping effectiveness at different temperatures with the same attached mass being used throughout. In both cases tested, the results are complicated by variability of firmness of attachment of the damper to the test plate as the double-coated Scotch tape responds to the temperature changes. It is clear, however, that temperature changes have less adverse effect upon dampers of fibrous glass materials than upon asphalted materials, despite the sharper tuning of the fibrous glass dampers.

Accuracy and Reproducibility of Measurements

The vibration decay rate measuring apparatus is capable of damping measurements of good accuracy and high reproducibility when the damping treatment under test is stable and itself physically reproducible. The accuracy of such measurements is estimated to be within ±2%, the error arising primarily in reading the decay rate photographs. In these experiments, however, reproducibility of results was limited by progressive physical changes in the damping treatment and dependence upon parameters hitherto neglected in decay rate measurements.

Though great care was taken from the outset of the experiments to prevent any damage to the respective spot dampers in changing them from one test plate to another, two other effects that could seriously affect the validity of decay rate measurements with spot dampers were not discovered until the experimental work was largely completed.

As it turned out, however, due to a carefully standardized measurement procedure, neither of these effects seriously impairs the relative accuracy of measurements taken at different frequencies.

It was noted during room-temperature measurements preceding the temperature tests of the asphalted felt spot damper that the damping effectiveness progressively increased with time during the first several hours immediately after fresh attachment of a damper to a test plate with new tabs of double-coated Scotch tape. In one case the increase amounted to 12% during the first two hours and 3% during the next twenty four hours. Fortunately, however, all of the measurements reported above were taken at approximately the same time after fresh attachment of the damper to a new test plate so these changes could not have seriously entered into the types of comparisons made above. The temperature tests, which required nearly 24 hours each to conduct, were not started until the attachment had been aged for several days and decay rates became accurately reproducible at room temperature.

Furthermore, the decay produced by a spot damper was found to be non-logarithmic to some extent, so that vibration decay rate measurements become somewhat dependent upon the level at which they are made. With an asphalted felt damper the vibration decay rates measured at very high level were found to be 10% greater than those measured at very low level. Though this figure represents an extreme case, some error due to this cause has undoubtedly entered into the measurements reported above.

Evaluation of the Results and Recommendations for Further Work

Pending the discovery of a damping material which incorporates the advantages of both the fibrous glass materials and the asphalted materials, the above measurements indicate that a spot damper can be fabricated which will satisfy each of the requirements for a practical damper for aircraft fuselage panels as listed in the contract for this work, insofar as follows: the damper weight will not exceed 0.15 pounds per square foot of panel treated, it will have a maximum damping action adjustable over a frequency range including 50-150 cps, it will provide a spread of very effective damping through a frequency range 50% below and at least 75% above the frequency of maximum damping, it will provide effective damping at higher frequencies at least up to 200 cps and most probably well beyond, its damping action will be effective at temperatures from -20 to +120 degrees F., and it will be resistant to fungus, vermin, and fire and will not contribute to the corrosion of aluminum. This listing implies a belief in the availability of such a composite material and the solution of the practical attachment problem without corrosive adhesives. It is recommended that future work emphasize first the survey of existing materials to find the optimum practical compromise between the various requirements, this phase to be followed by the more exacting work of determining how tuning depends upon the various physical parameters of the damper. Existing equipment will be entirely adequate for a materials survey, but further refinement may be required for the basic study of the damping mechanism, especially if the frequency range of investigation is to be extended.

APPENDIX I

Correction of Vibration Decay Rate For Differences in Test Plate Masses

The dynamic equivalence demonstrated by the geometrical similarity of the vibration patterns of the various test plates permits an idealization of each of the test plates to its equivalent one-degree-of-freedom lumped parameter system. The errors introduced by such idealization are the same for each test plate, to first approximation.

The effective vibrating mass of each test plate is some fraction "a" of its total actual mass, hence

$$m_i = aM_i$$

when m_i = the effective vibrating mass of the i^{th} plate, M_i = the actual total mass of the i^{th} plate, and "a" is the same for all plates because of their dynamic equivalence. Since the undamped natural frequency ω_0 of each plate is known experimentally, the effective stiffness of each plate is determined by the relation

$$(\omega_0)_i = \sqrt{k_i/m_i}$$

so that the critical damping constant of each plate can be defined as is usual by

$$(c_c)_i = 2\sqrt{k_i m_i} .$$

Denoting by D_i the vibration decay rate (db/sec) measured on the i^{th} plate of a damping treatment represented by the lumped parameter c_i, the relation referred to in the text becomes

$$D_i = b(\omega_0)_i \frac{c_i}{(c_c)_i}$$

where "b" is a constant independent of the plate used. Hence a vibration decay rate measured with the treatment attached to the i^{th} plate depends not only upon the damping of the treatment which is characterized here by c_i but also upon the critical damping constant of the plate, $(c_c)_i$. In order to remove the dependence of the vibration decay rate upon the plate with which it was measured, a correction must be devised for each vibration decay rate measurement which depends only upon known parameters of the plate.

Rearranging the above relation between decay rate and fraction of critical damping,

$$c_i = \frac{1}{b(\omega_o)_i} (c_c)_i \, D_i .$$

Since the damping of the treatment under test must be independent of the test plate which is used,

$$\frac{1}{(\omega_o)_i}(c_c)_i \, D_i = \frac{1}{(\omega_o)_j}(c_c)_j \, D_j ,$$

$$\frac{D_i}{D_j} = \frac{(\omega_o)_i (c_c)_j}{(\omega_o)_j (c_c)_i} .$$

Using the expressions for $(\omega_o)_i$ and $(c_c)_i$ from above, however, reduces the relations to

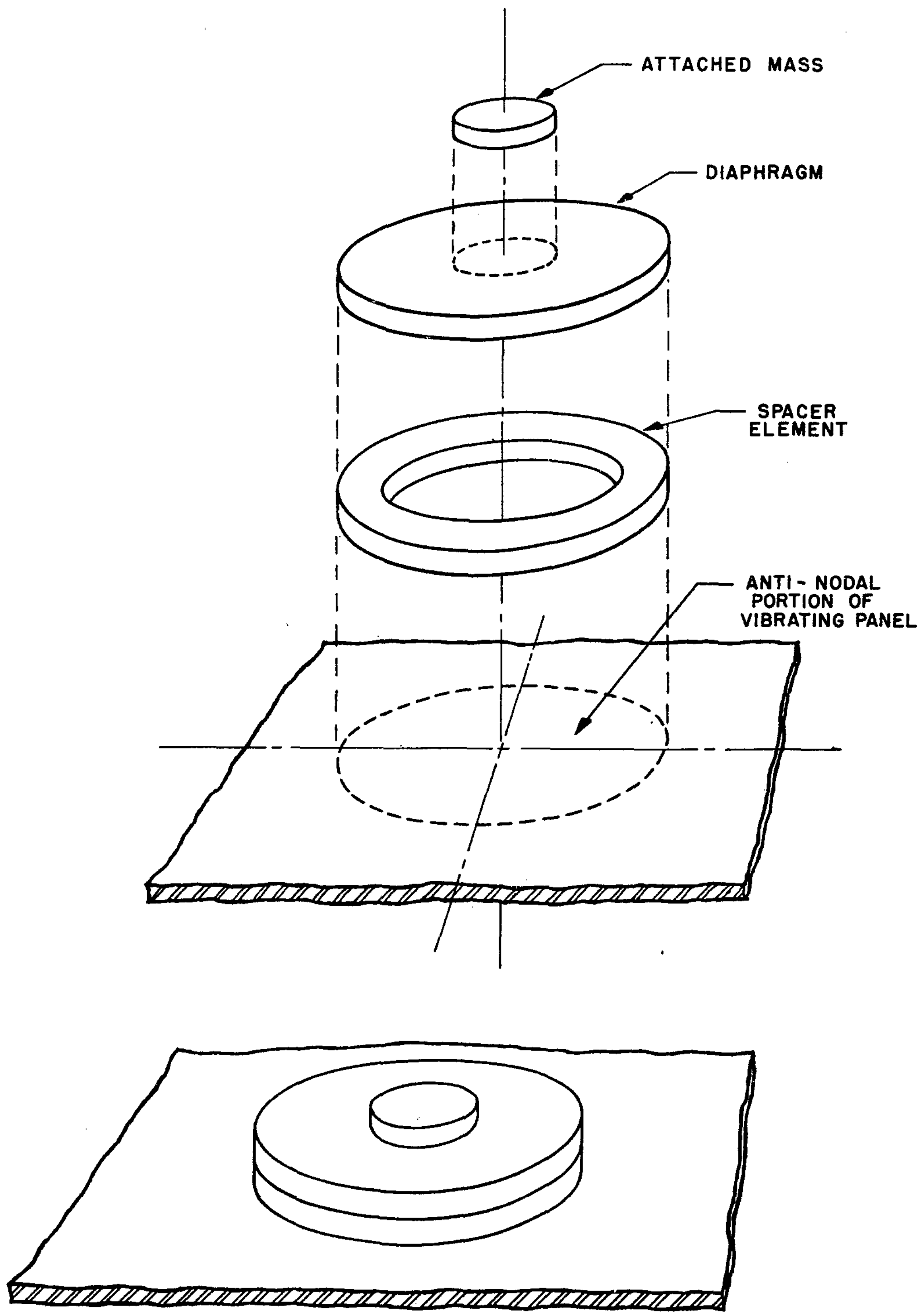
$$\frac{D_i}{D_j} = \frac{m_j}{m_i} = \frac{aM_j}{aM_i} = \frac{M_j}{M_i}$$

$$D_i = \frac{M_j}{M_i} D_j .$$

Hence, a comparison of the damping effectiveness on the basis of vibration decay rate measurements made with different test plates requires the adjustment of the decay rates of one plate with respect to the other by a factor involving the masses of the respective test plates. In these experiments the 180 cps test plate was regarded as the standard system against which to compare all other values, hence all vibration decay rates have been adjusted by the equation

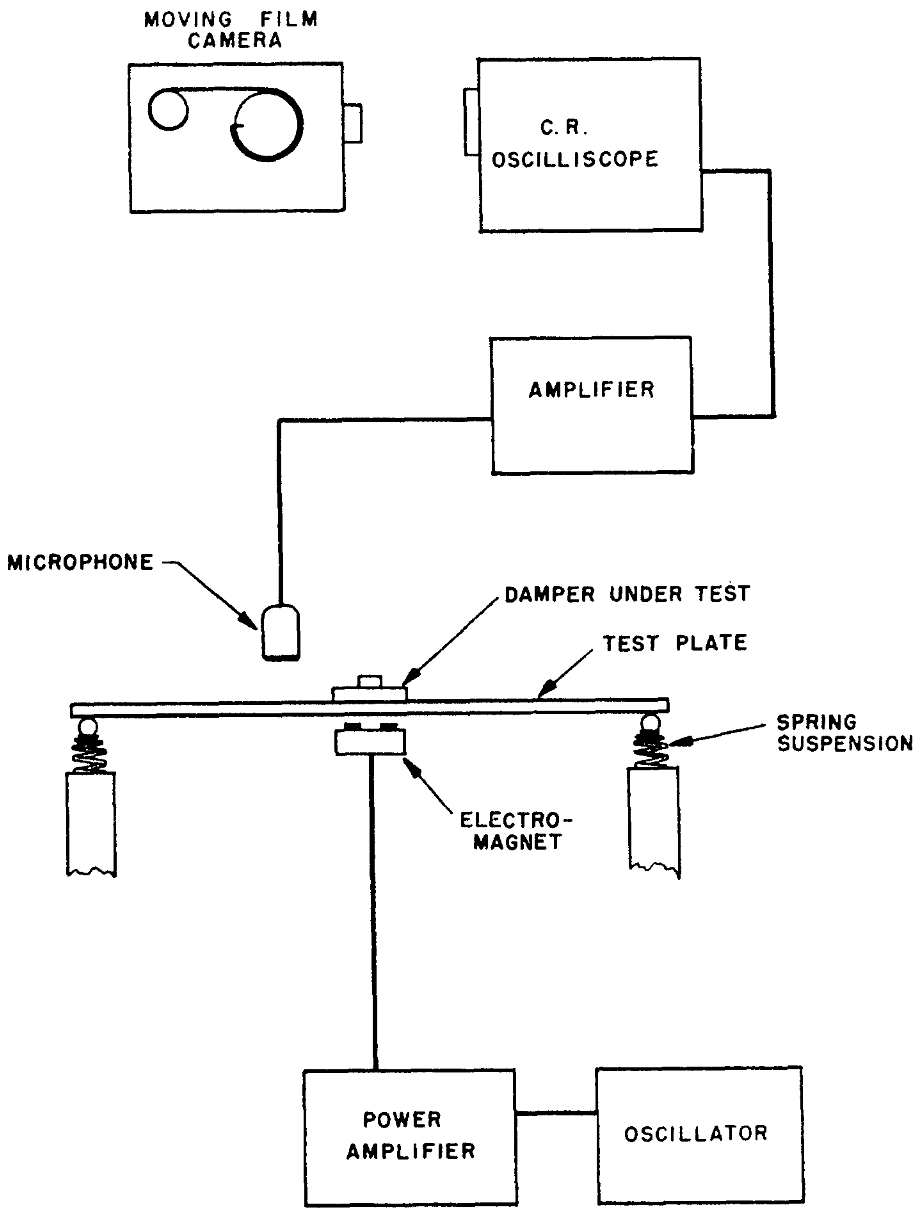
$$D_{corrected} = \frac{M_i}{M_{180}} D_{measured}$$

where M_i is the total weight of the test plate on which the decay rate was measured and M_{180} is the total weight of the 180 cps test plate.



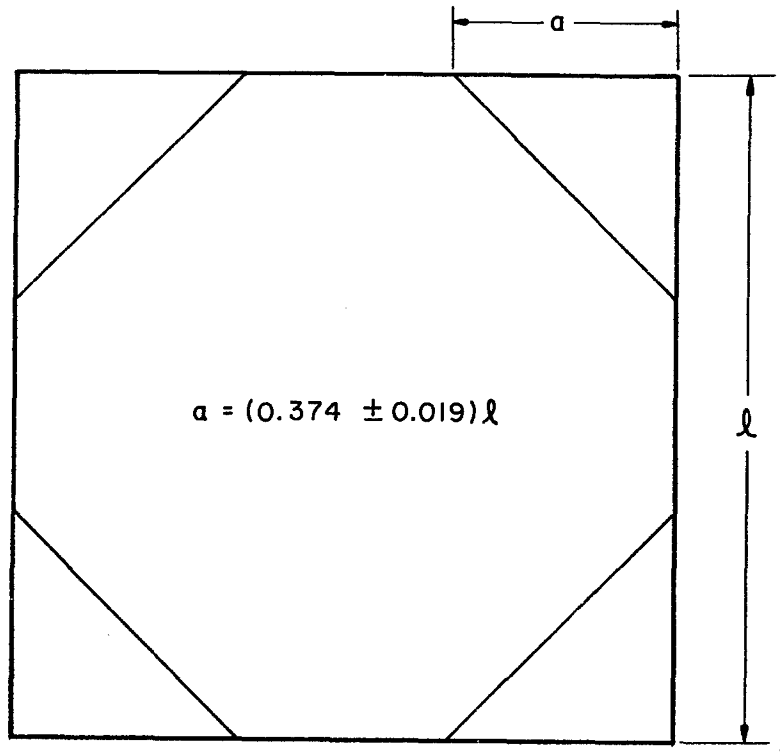
SCHEMATIC DIAGRAM OF SPOT DAMPER

FIG. 2



SCHEMATIC DIAGRAM OF EXPERIMENTAL APPARATUS

FIG. 3



VIBRATION PATTERN OF TEST PLATES

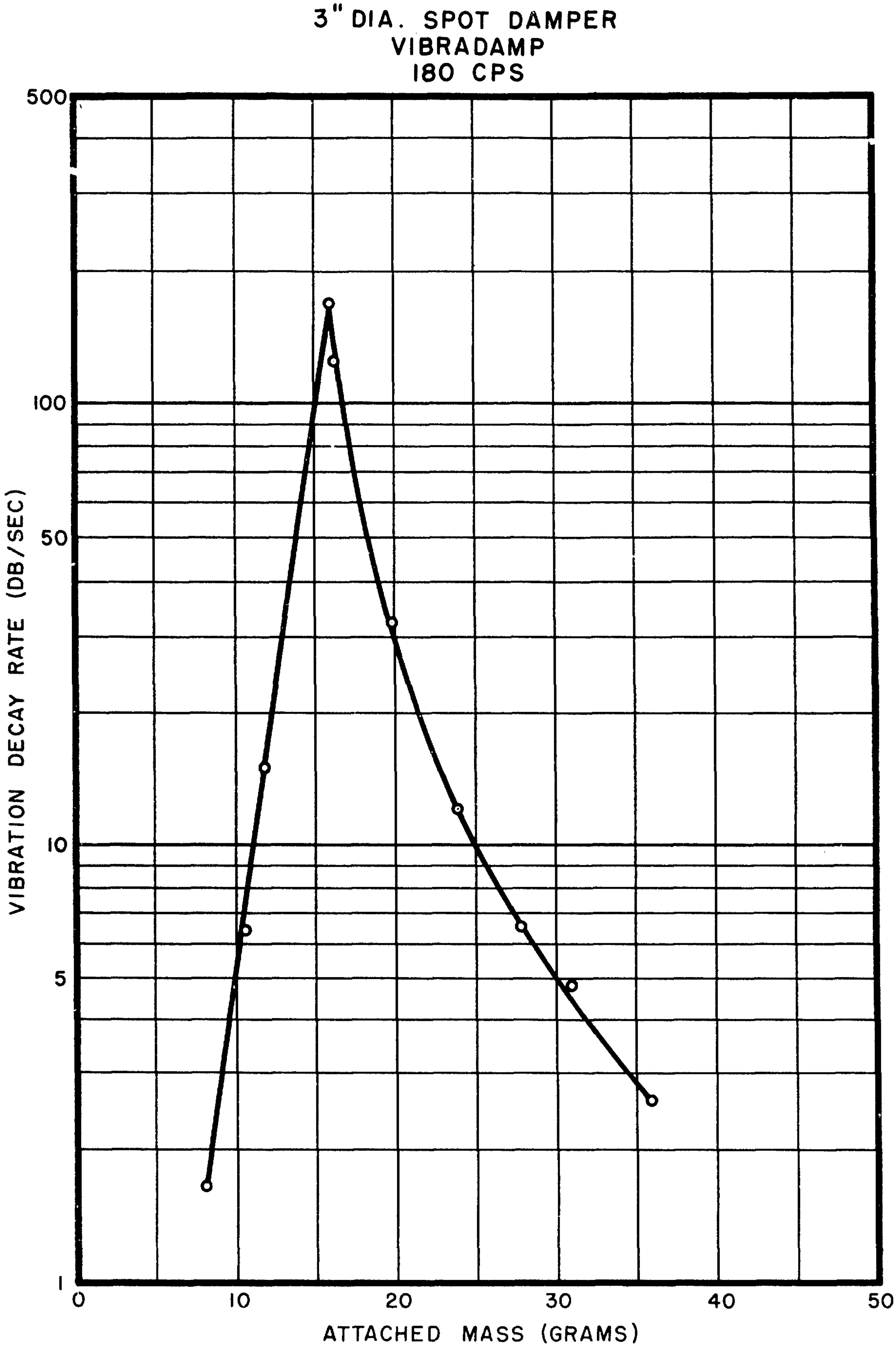
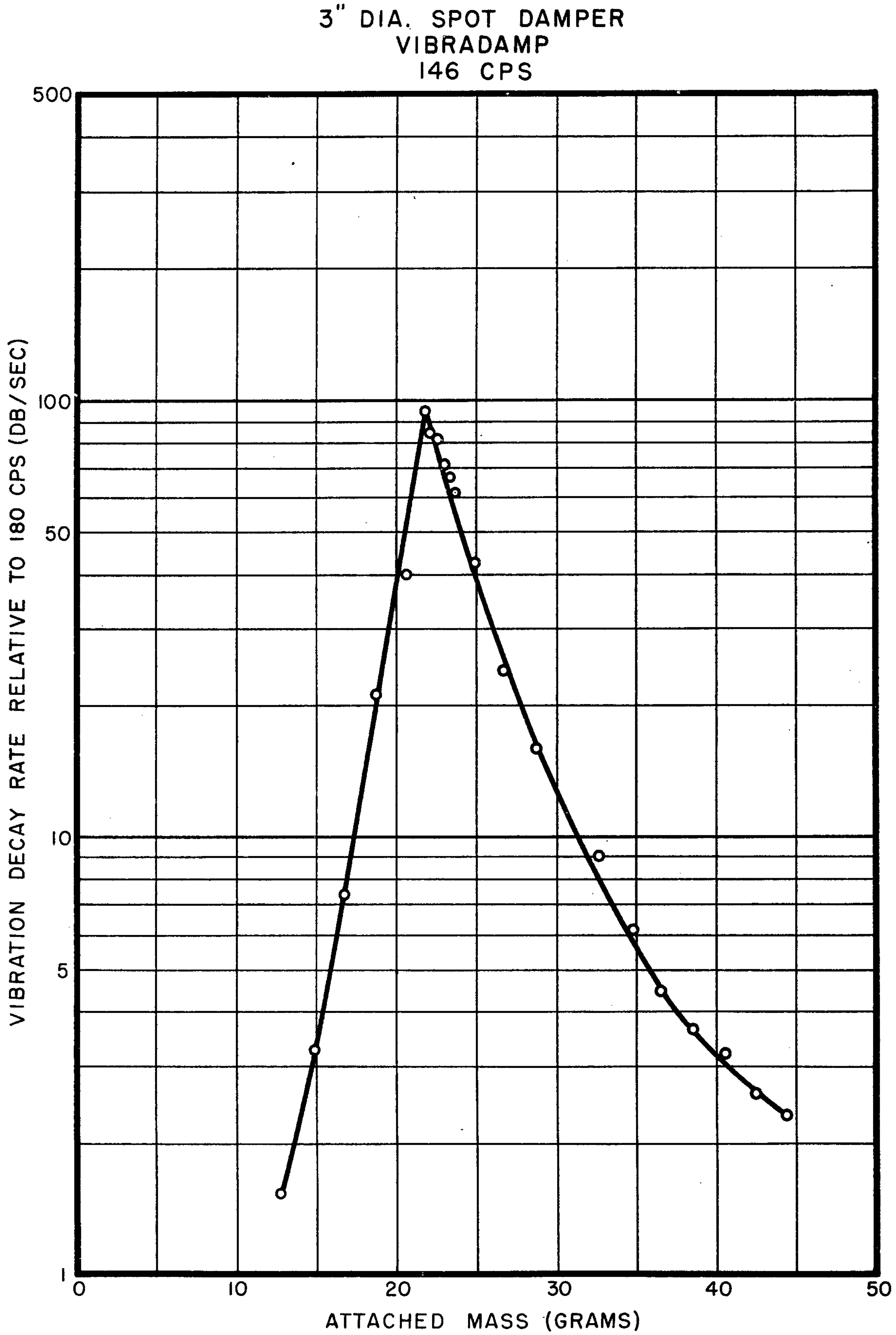

3" DIA. SPOT DAMPER
VIBRADAMP
180 CPS
500
100
50
10
5
1
VIBRATION DECAY RATE (DB/SEC)
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

FIG. 5


3" DIA. SPOT DAMPER
VIBRADAMP
146 CPS
500
100
50
10
5
1
0
10
20
30
40
50
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
ATTACHED MASS (GRAMS)

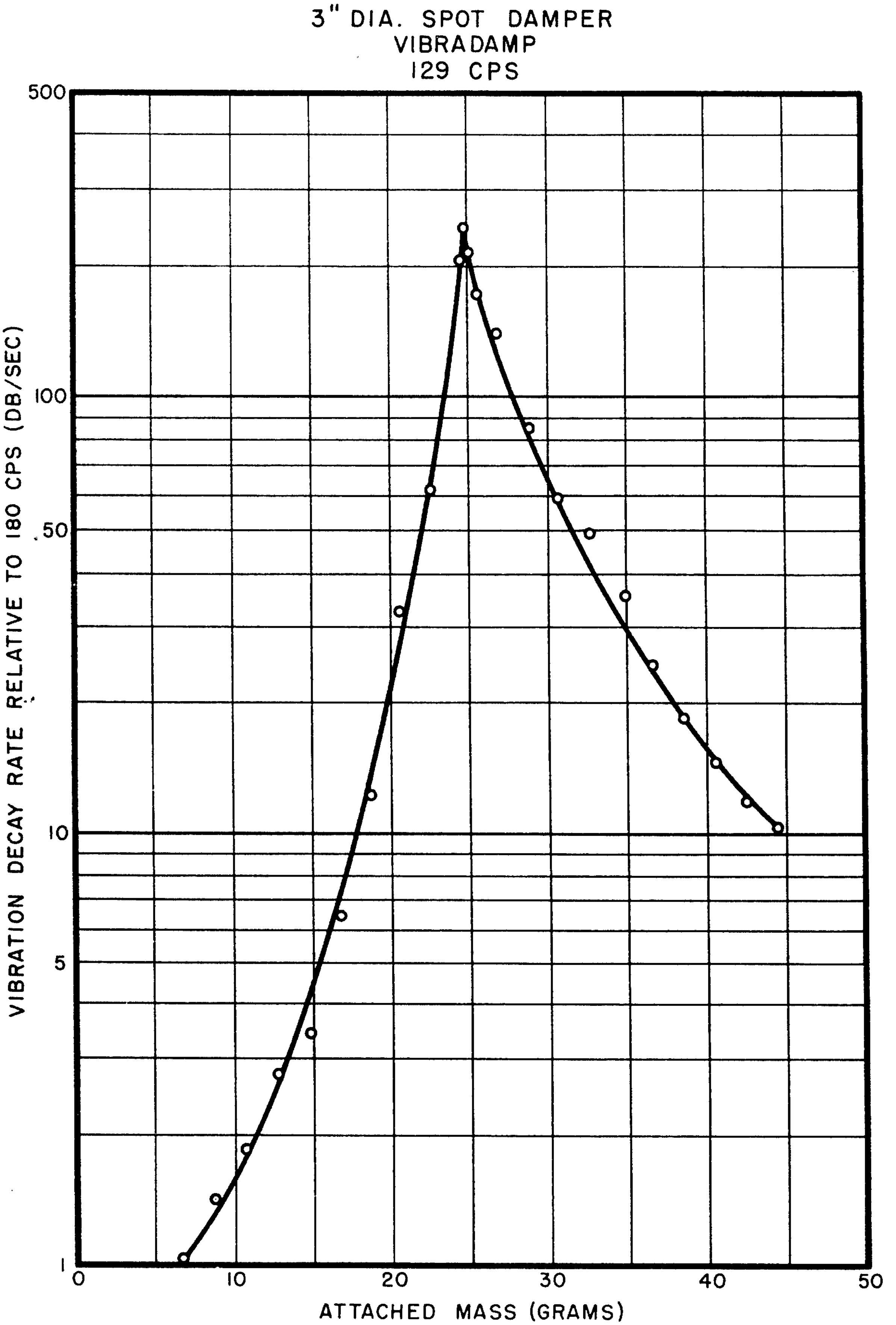
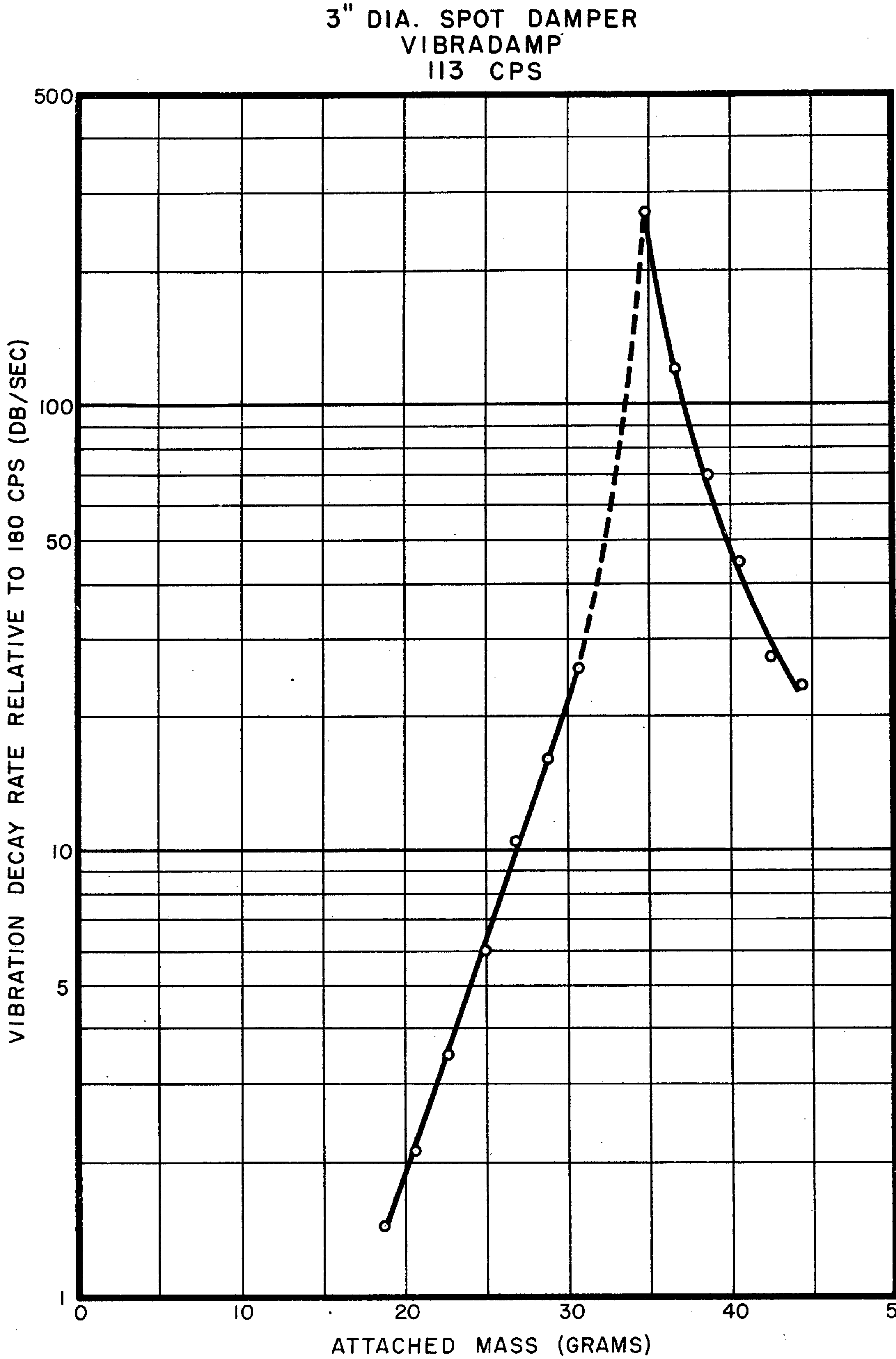
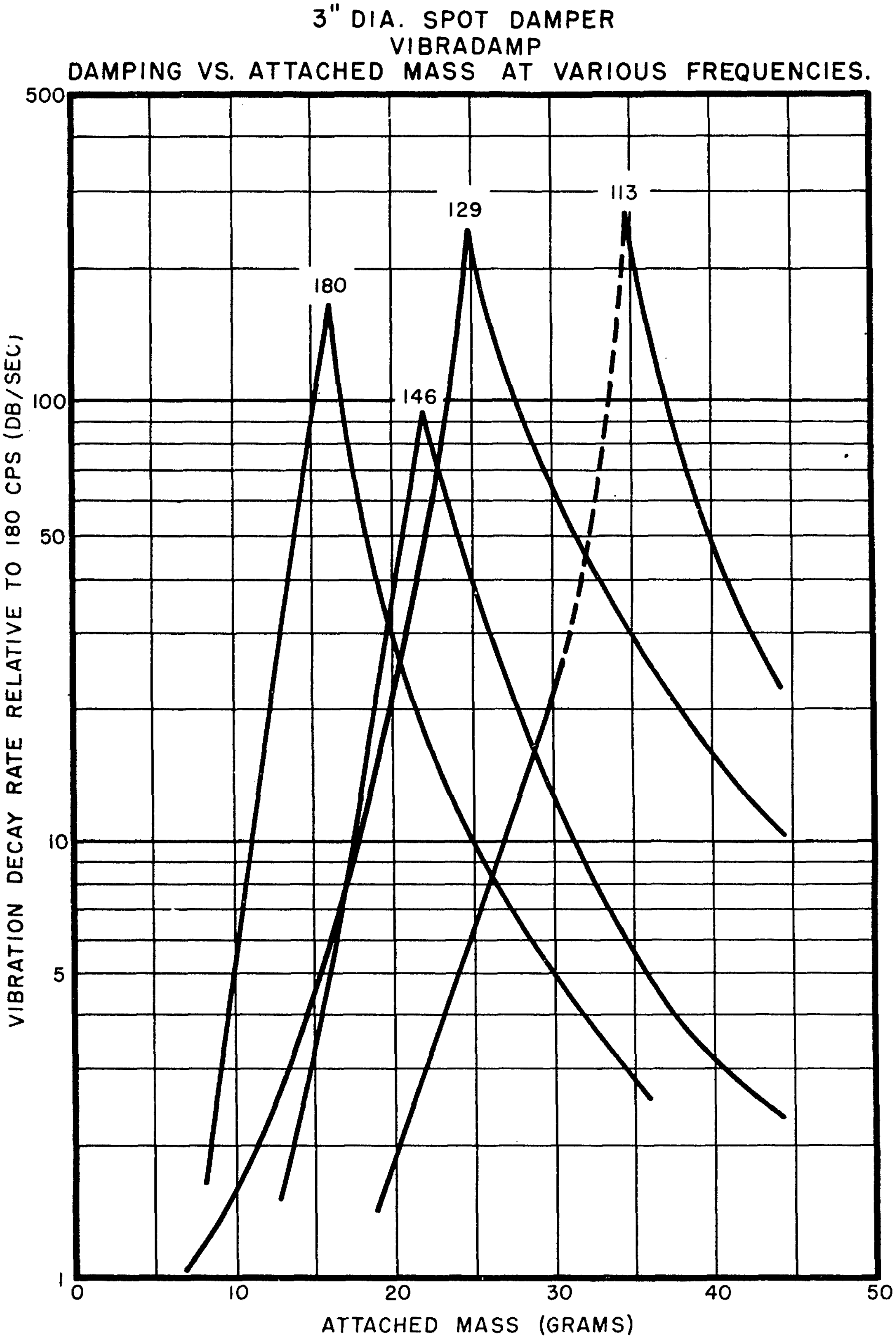

3" DIA. SPOT DAMPER
VIBRADAMP
129 CPS
500
100
50
10
5
1
0
10
20
30
40
50
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
ATTACHED MASS (GRAMS)

FIG. 7



3" DIA. SPOT DAMPER
VIBRADAMP
113 CPS
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
500
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

FIG. 8



3" DIA. SPOT DAMPER
VIBRADAMP
DAMPING VS. ATTACHED MASS AT VARIOUS FREQUENCIES.
500
100
50
10
5
1
0
10
20
30
40
50
180
146
129
113
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
ATTACHED MASS (GRAMS)

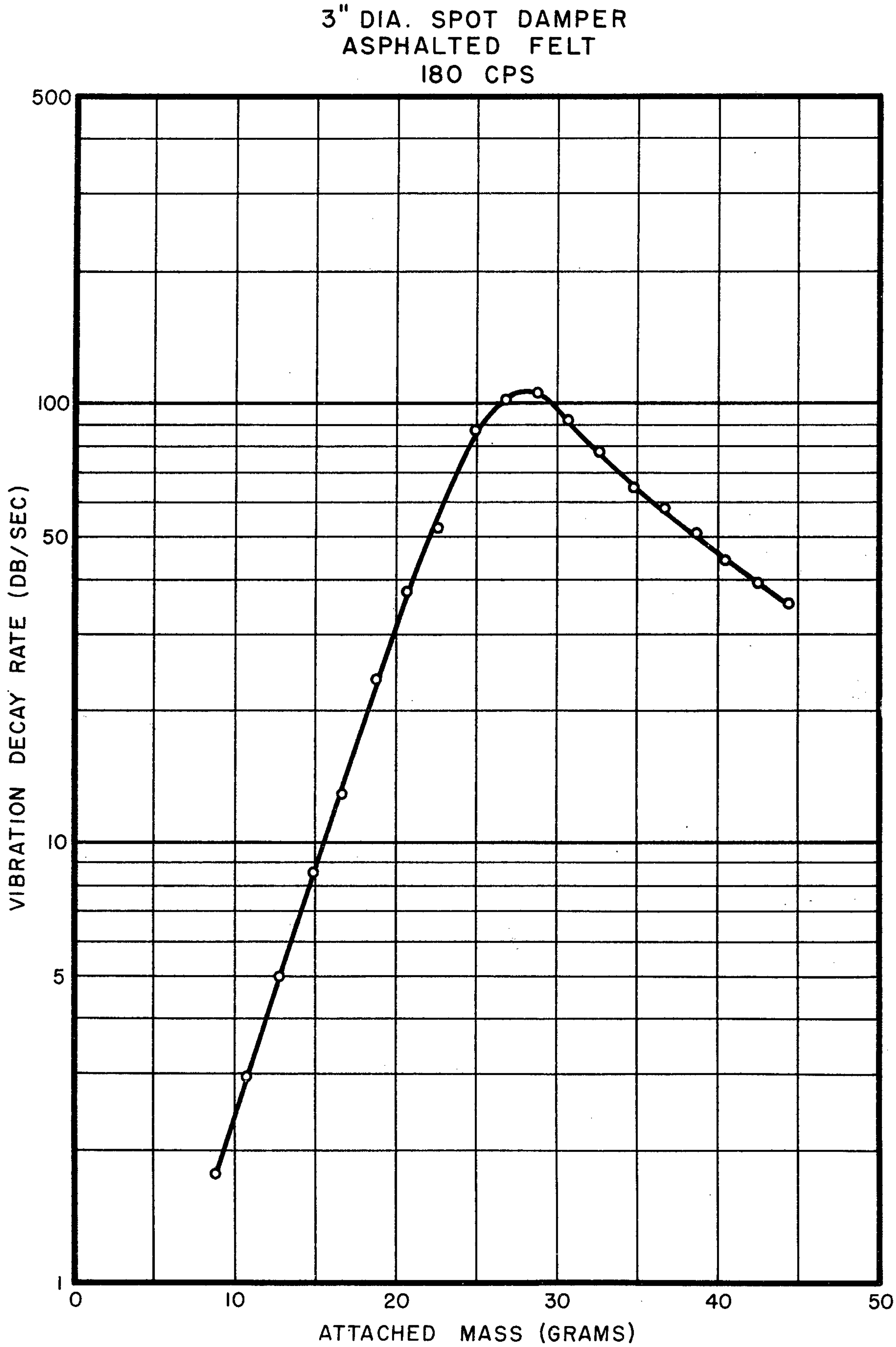
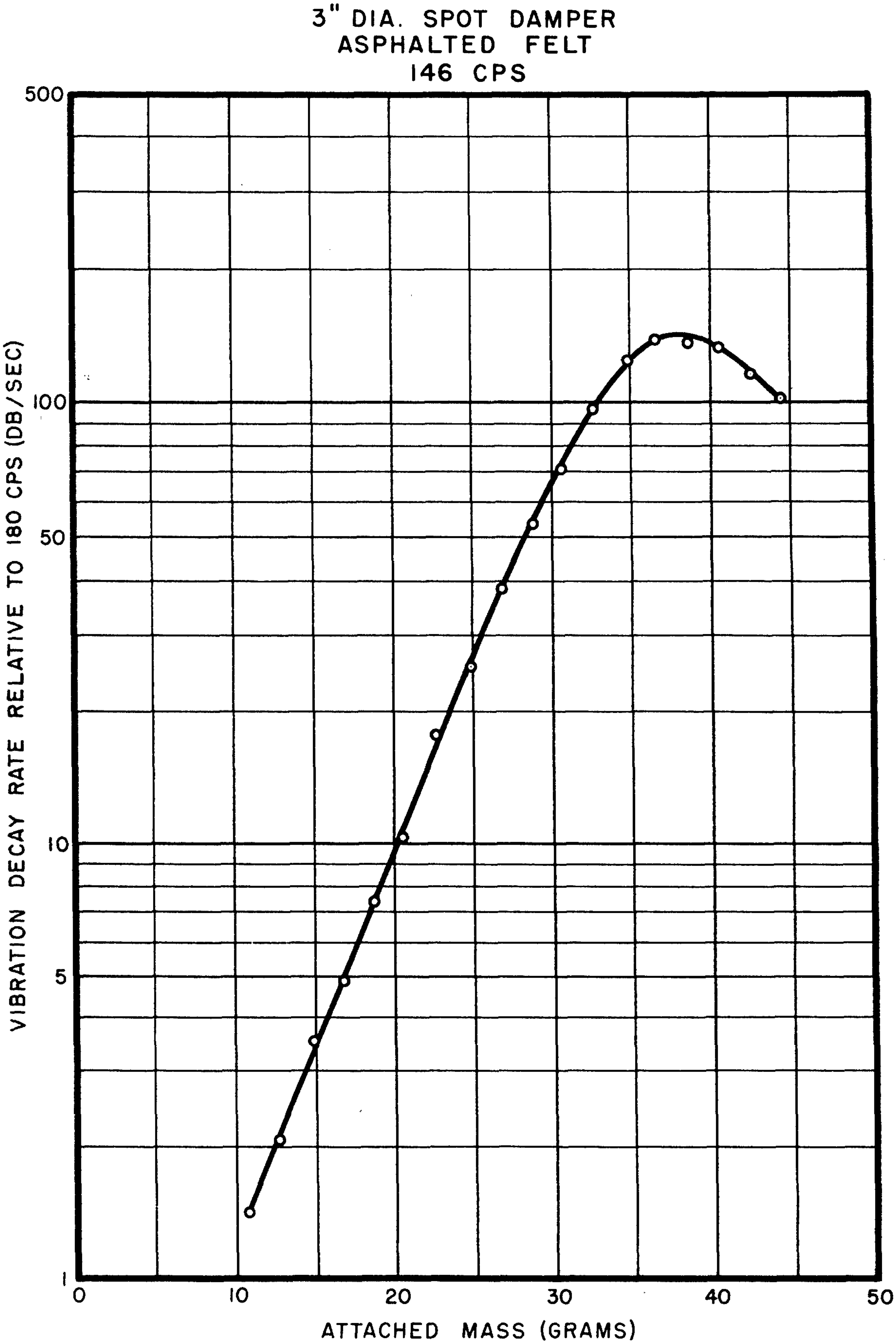
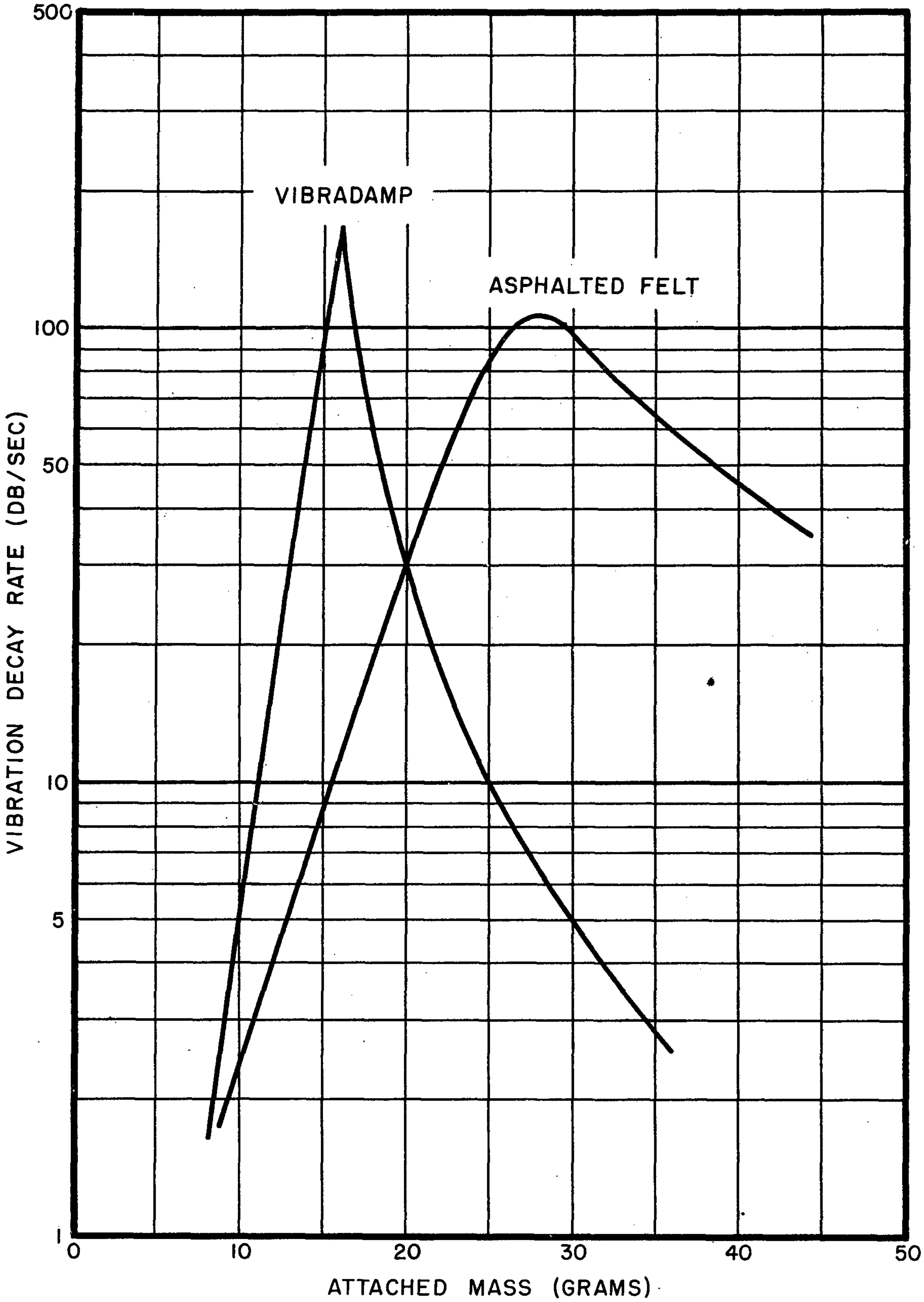

3" DIA. SPOT DAMPER
ASPHALTED FELT
180 CPS
500
100
50
10
5
1
0
10
20
30
40
50
VIBRATION DECAY RATE (DB/SEC)
ATTACHED MASS (GRAMS)

FIG. 10



3" DIA. SPOT DAMPER
ASPHALTED FELT
146 CPS
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
500
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

3" DIA. SPOT DAMPERS
COMPARISON OF VIBRADAMP AND ASPHALTED FELT
180 CPS



COMPARISON OF TWO DAMPERS OF DIFFERENT DIA.
ASPHALTED FELT
180 CPS

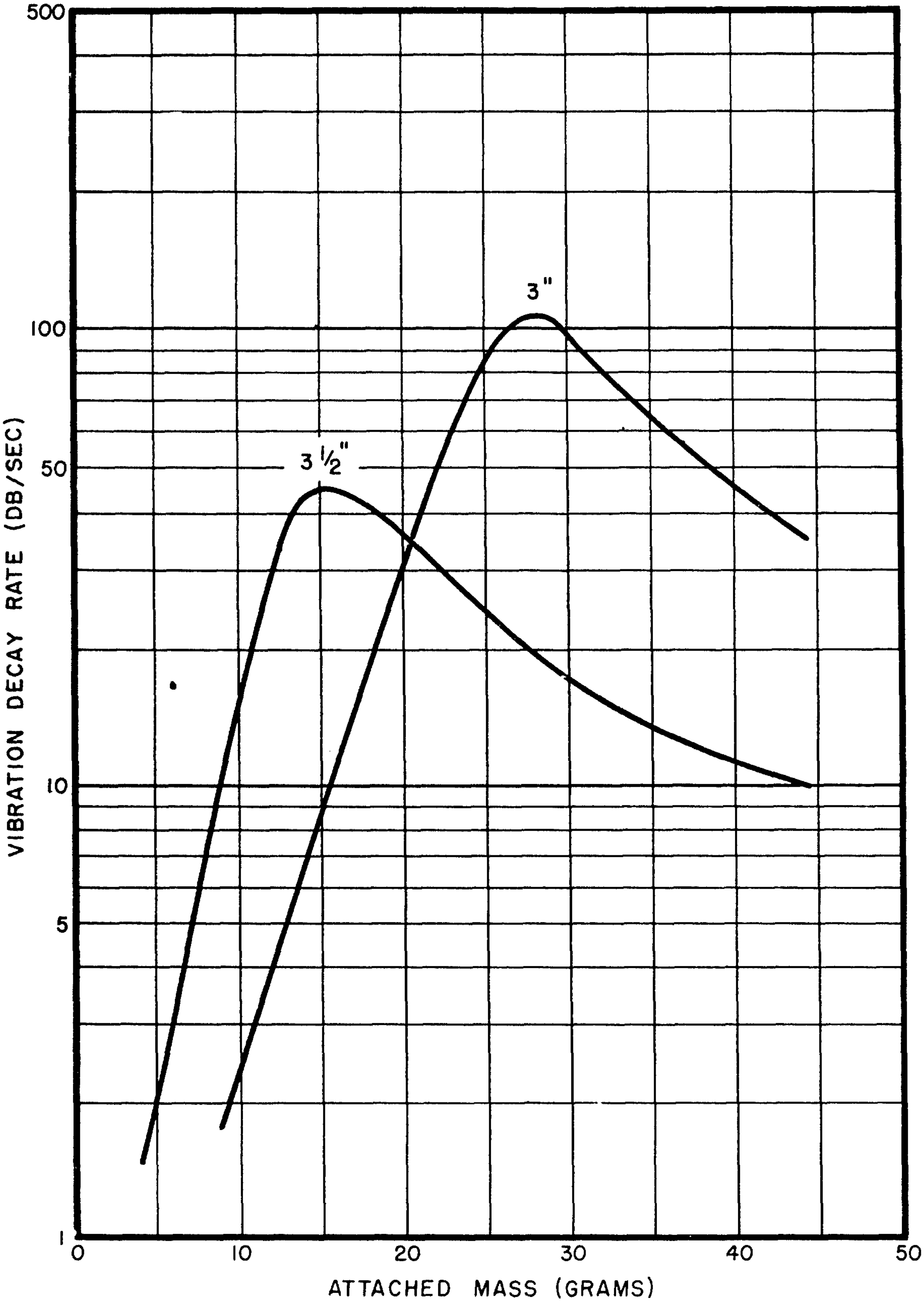
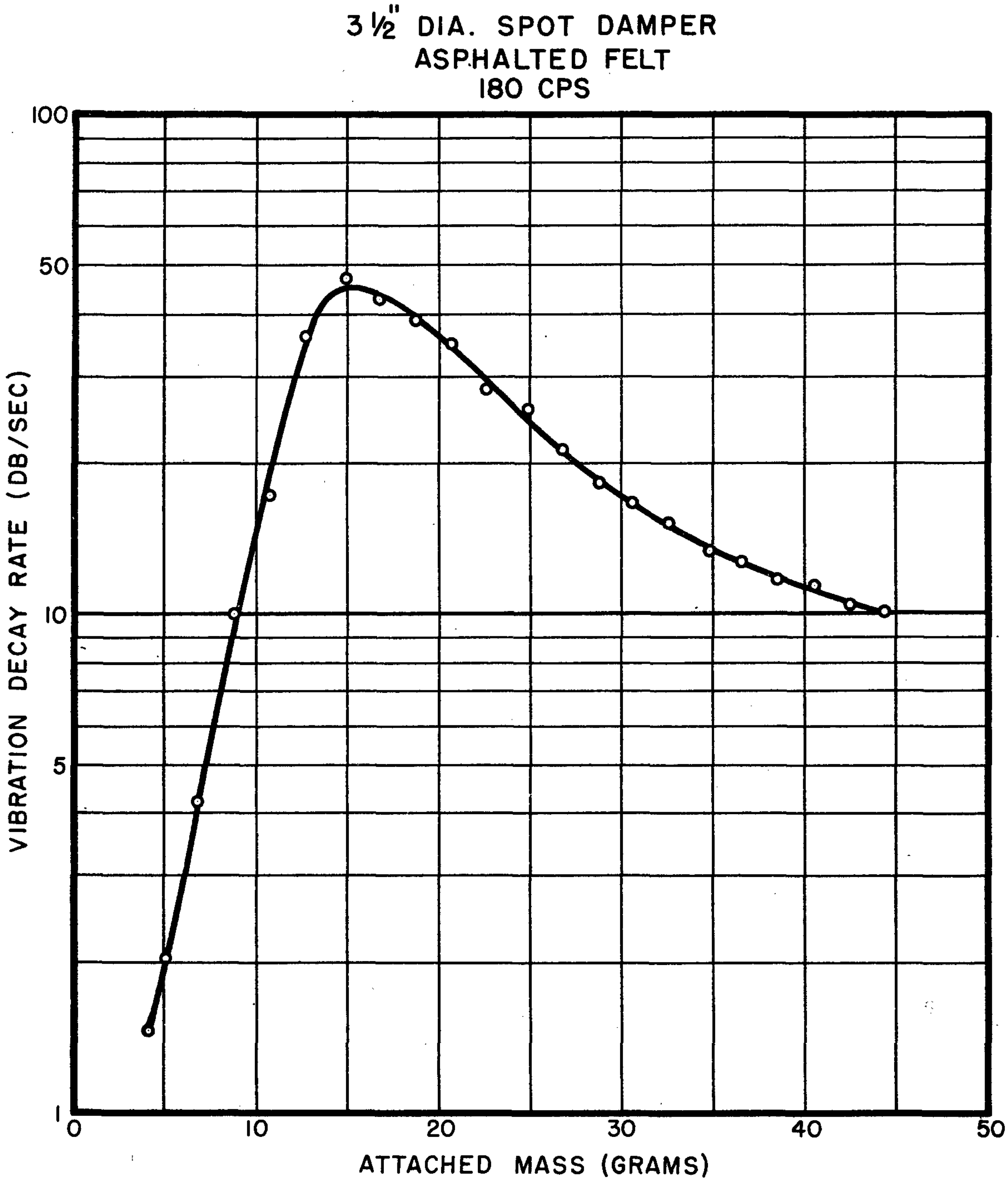
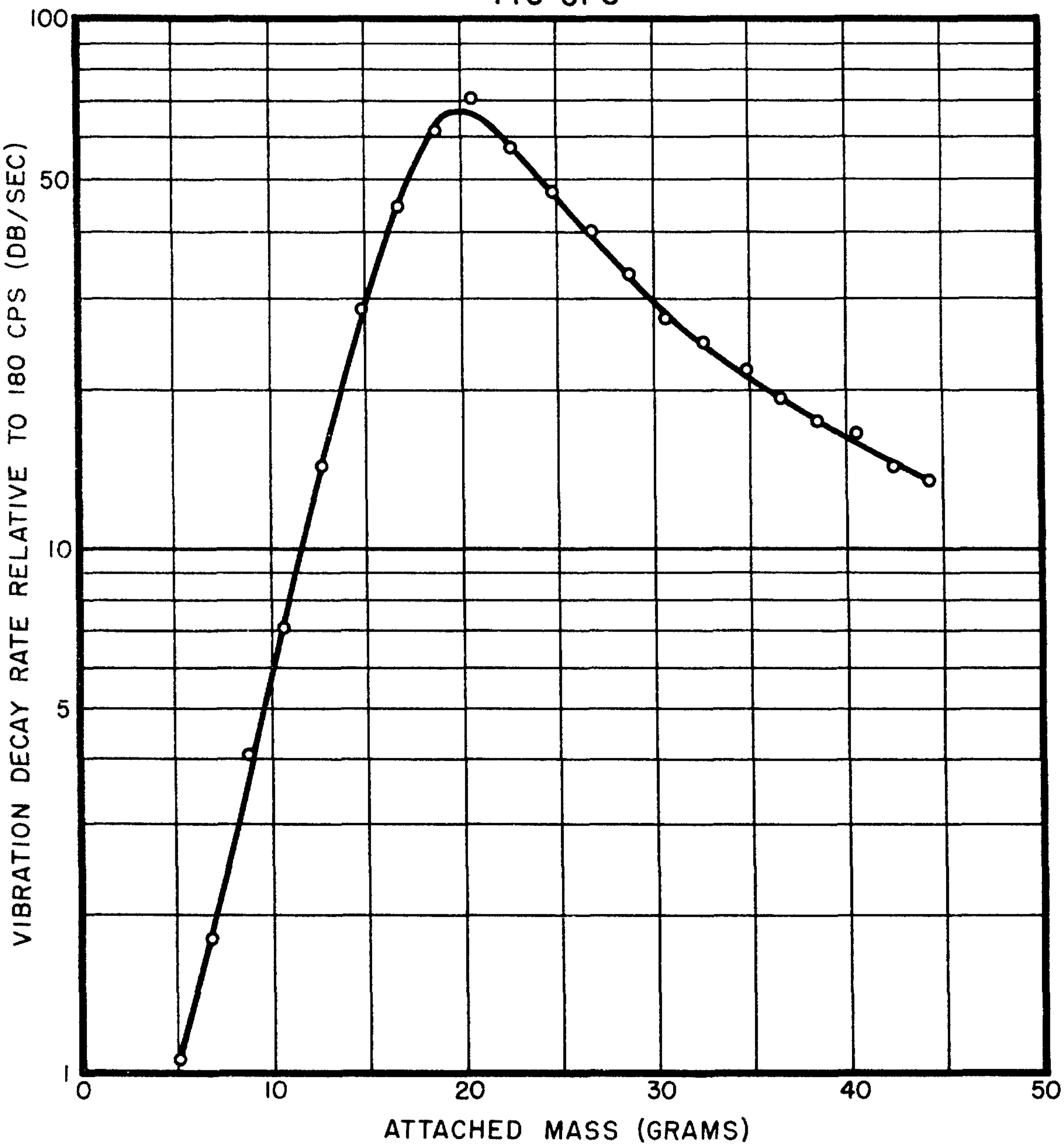


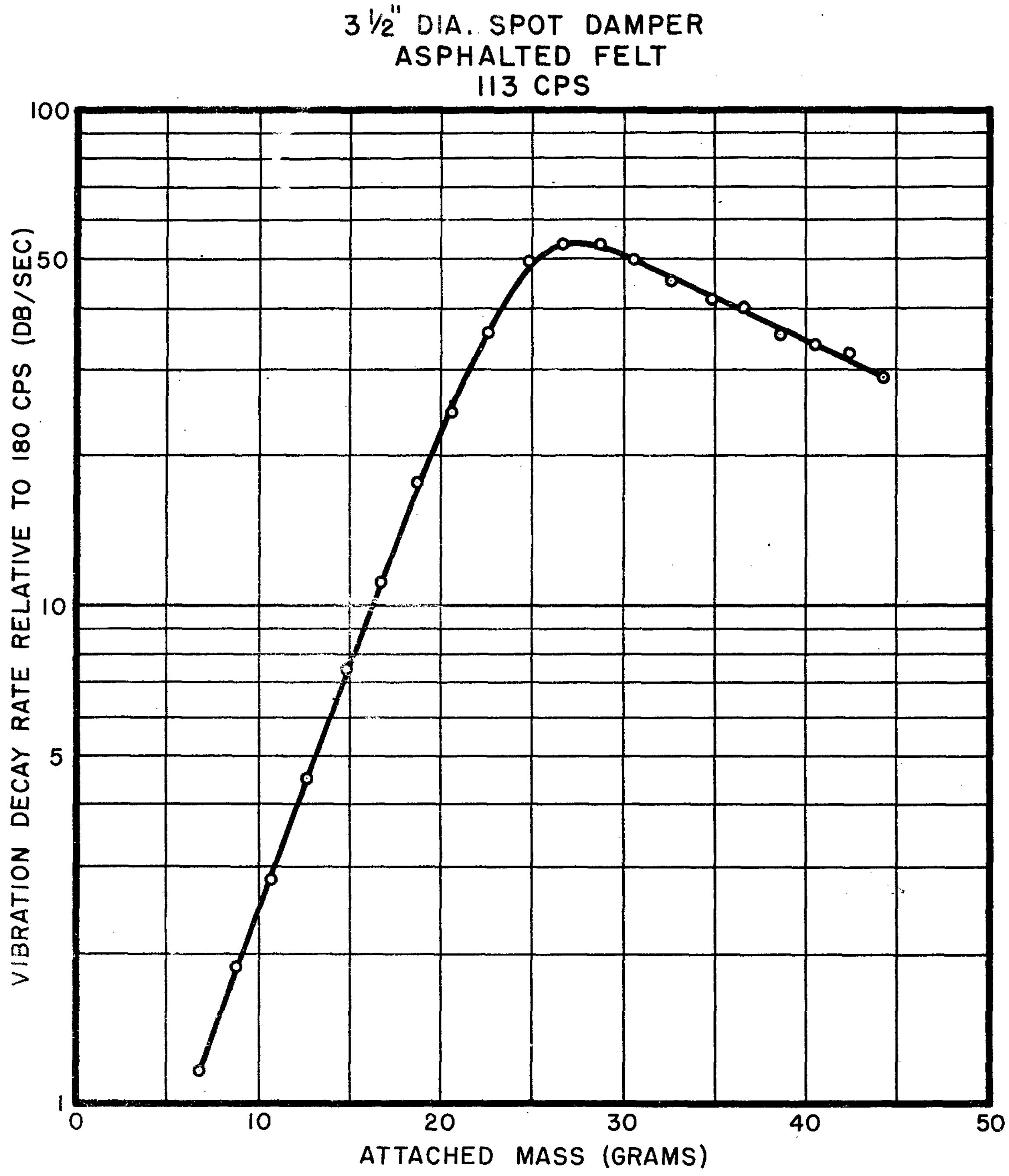
FIG. 13

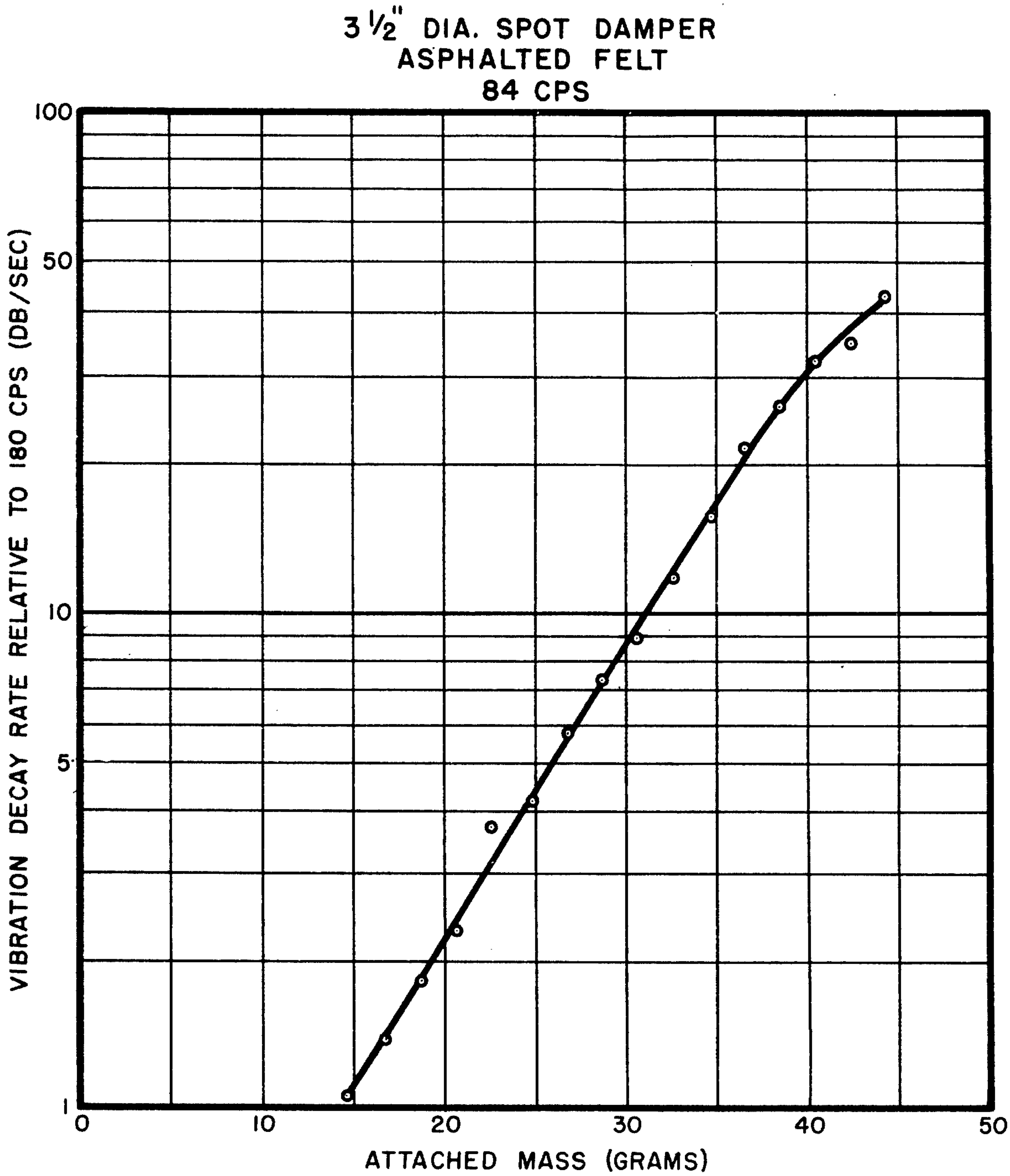


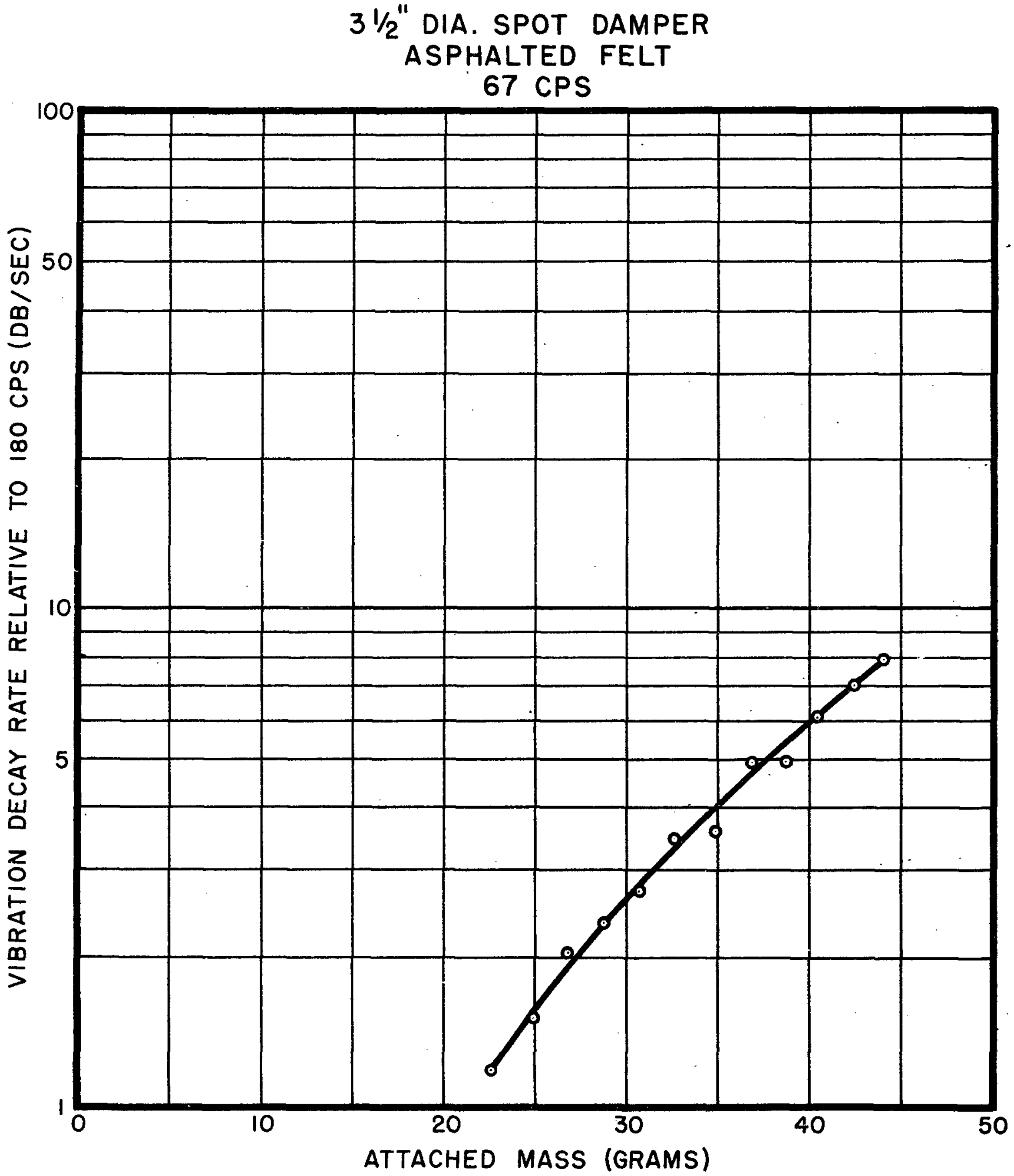
3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
180 CPS
VIBRATION DECAY RATE (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
146 CPS

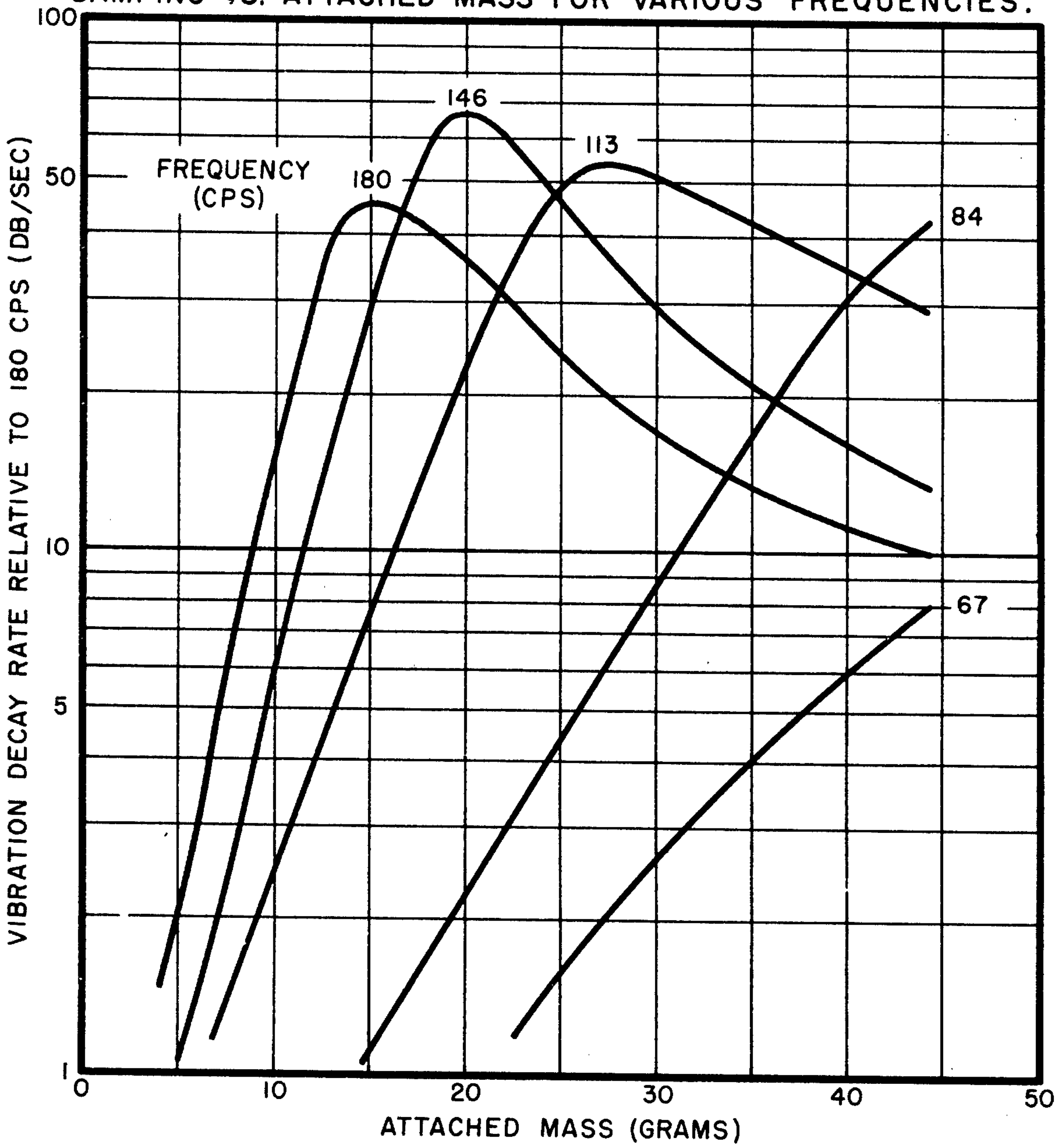



3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
113 CPS
100
50
10
5
1
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
0
10
20
30
40
50
ATTACHED MASS (GRAMS)


3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
84 CPS
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)


3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
67 CPS
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
DAMPING VS. ATTACHED MASS FOR VARIOUS FREQUENCIES.



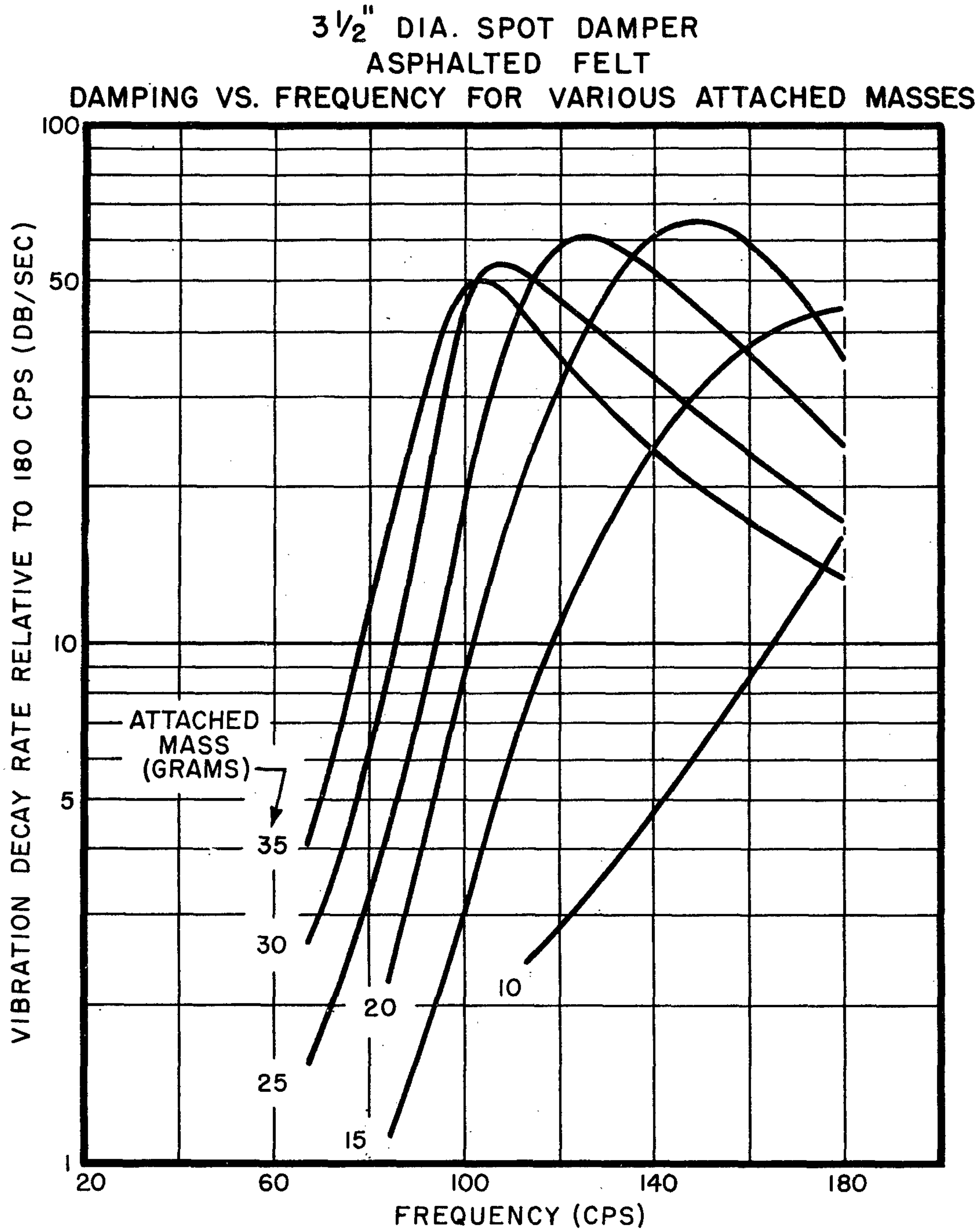
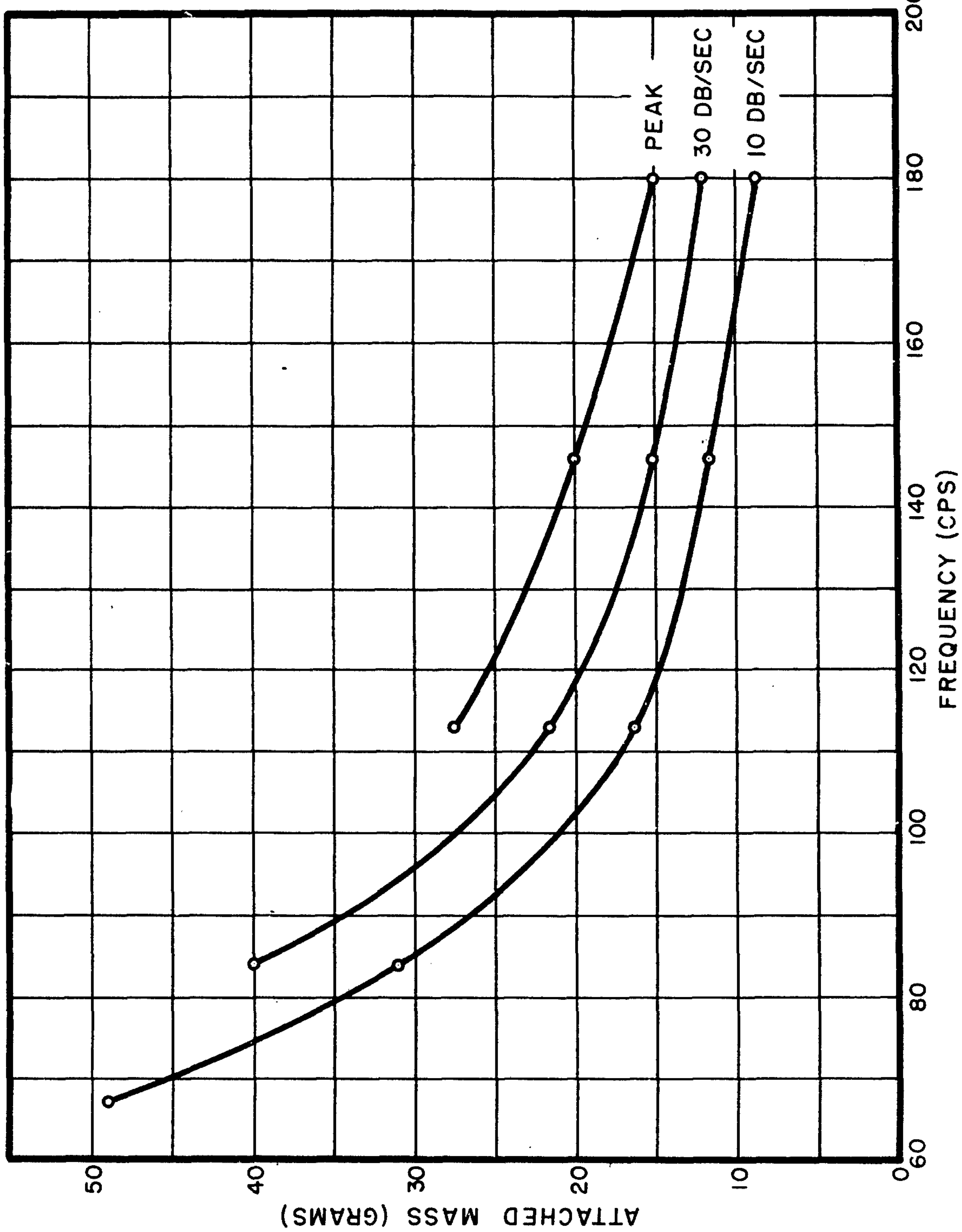
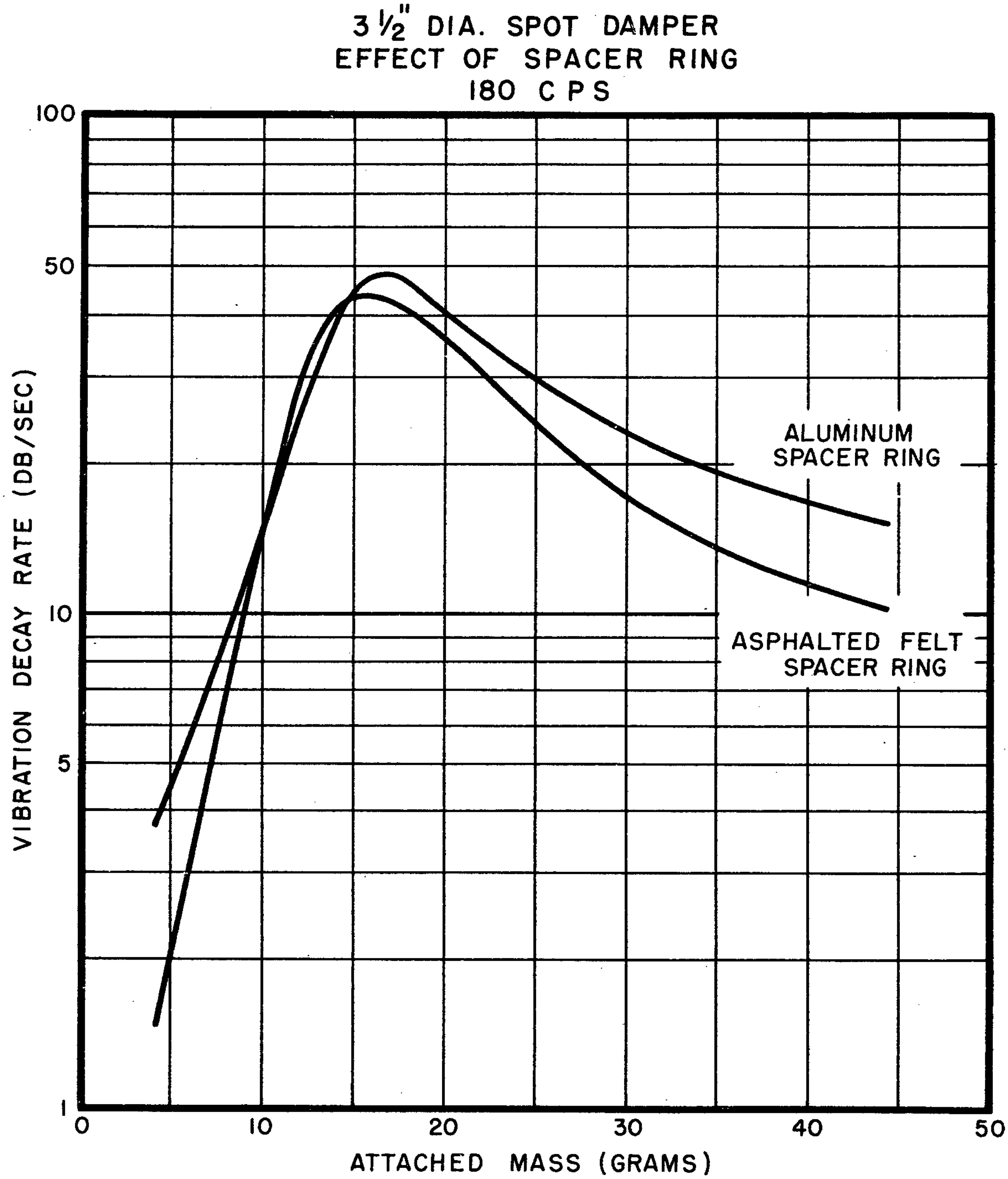
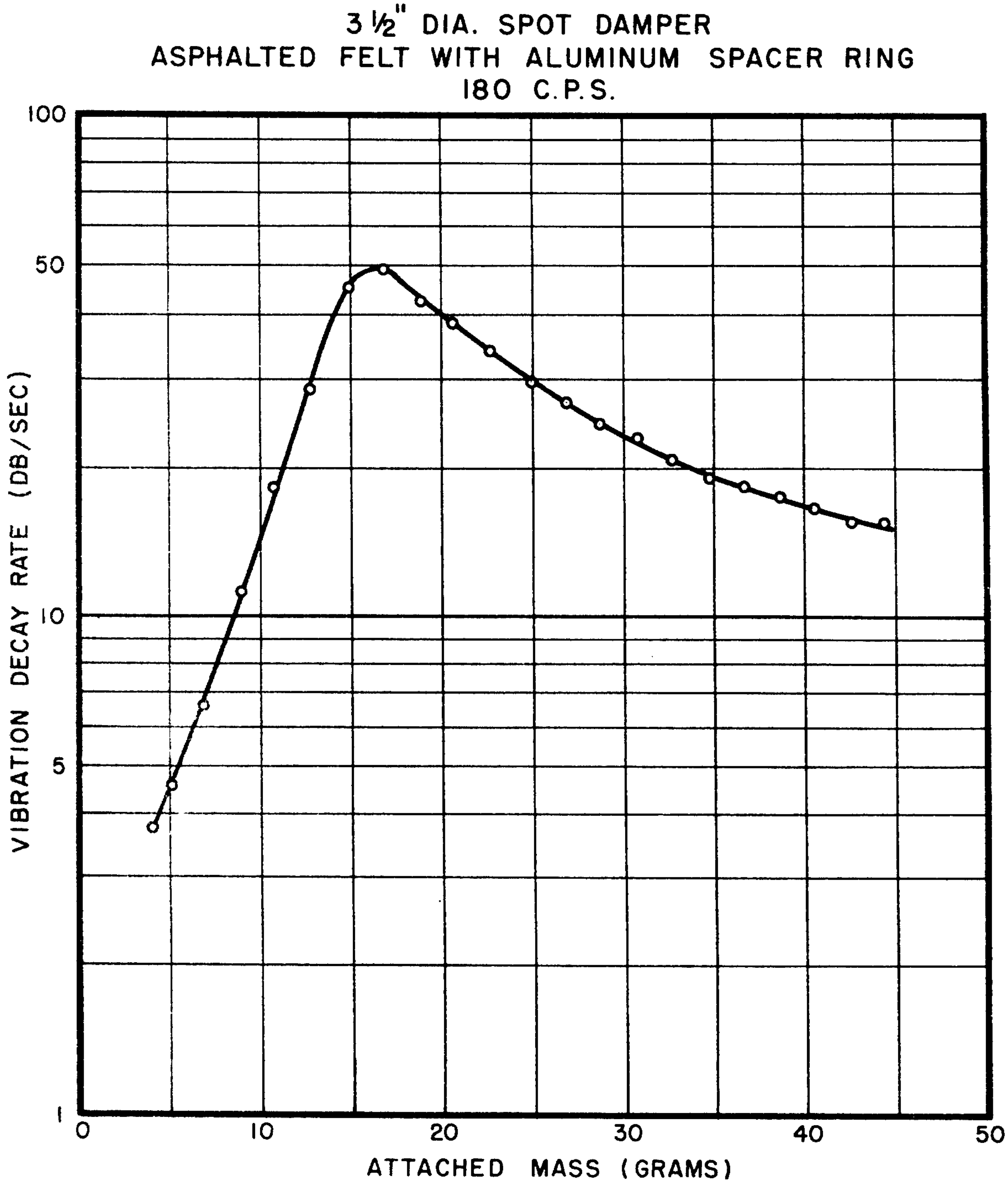

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
DAMPING VS. FREQUENCY FOR VARIOUS ATTACHED MASSES
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
100
50
10
5
1
ATTACHED MASS (GRAMS)
35
30
25
20
15
10
20
60
100
140
180
FREQUENCY (CPS)

FIG. 20

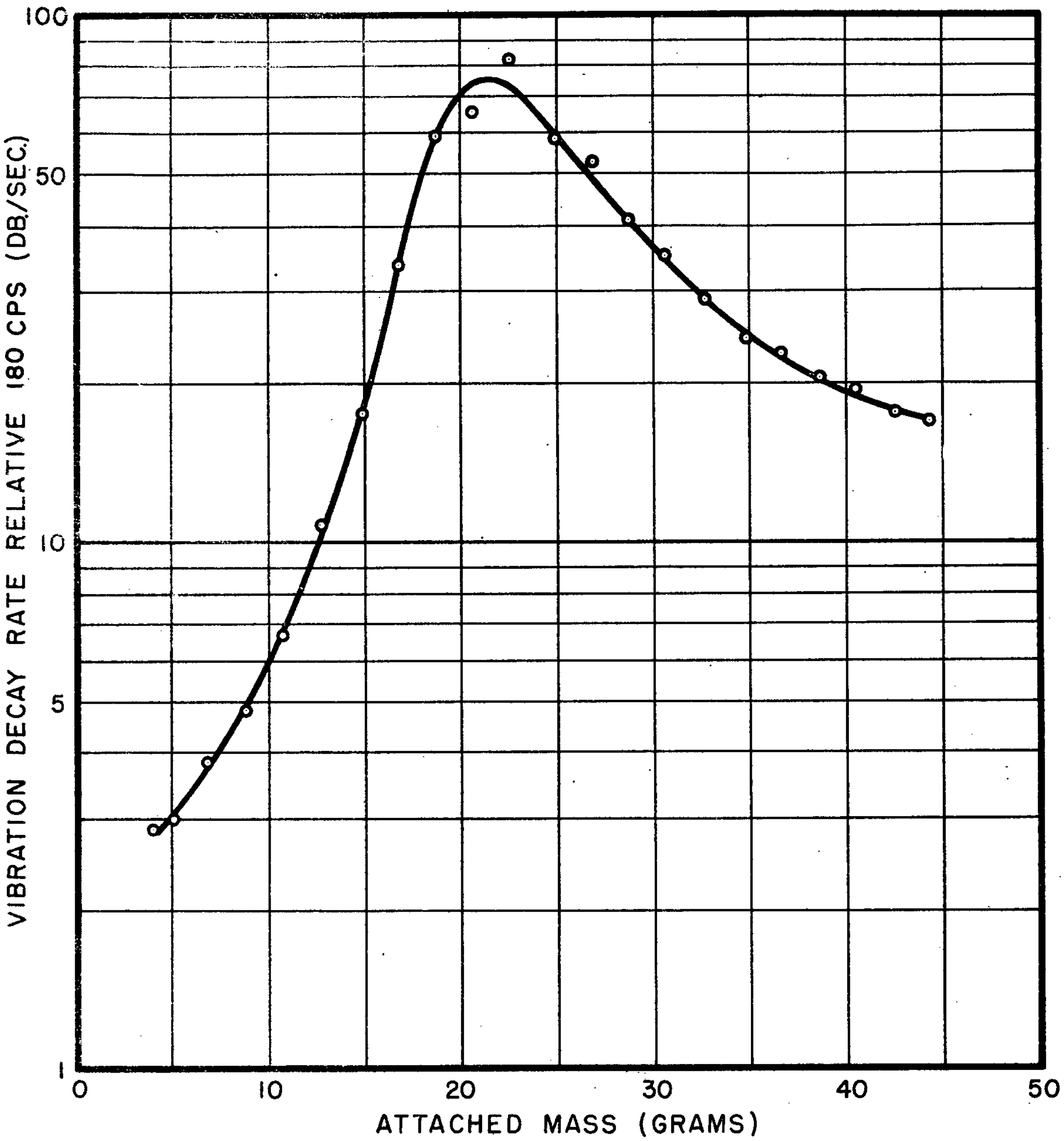
3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT
ATTACHED MASS VS. FREQUENCY FOR VARIOUS AMOUNTS OF DAMPING.

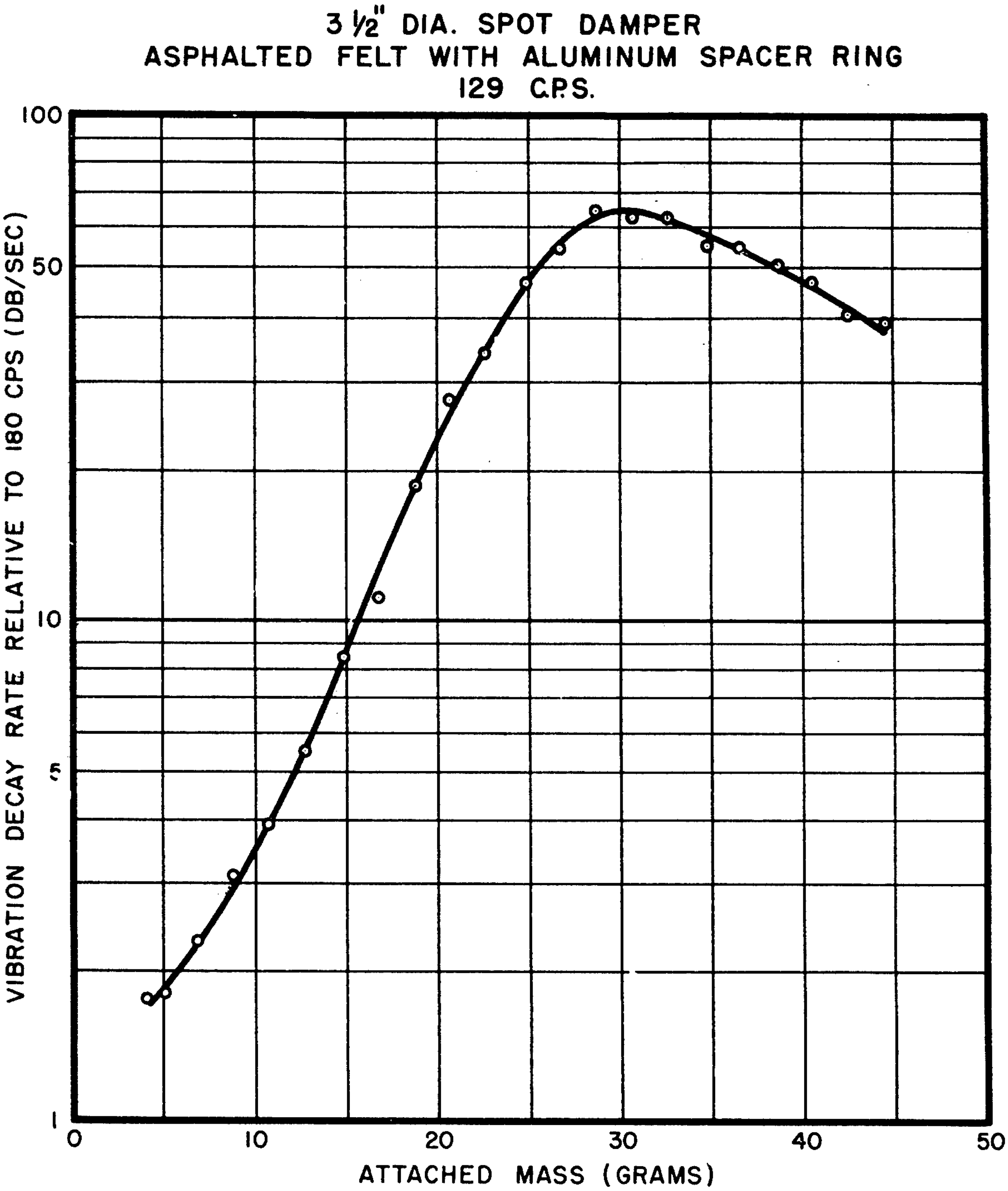



3 1/2" DIA. SPOT DAMPER
EFFECT OF SPACER RING
180 CPS
VIBRATION DECAY RATE (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)
ALUMINUM SPACER RING
ASPHALTED FELT SPACER RING

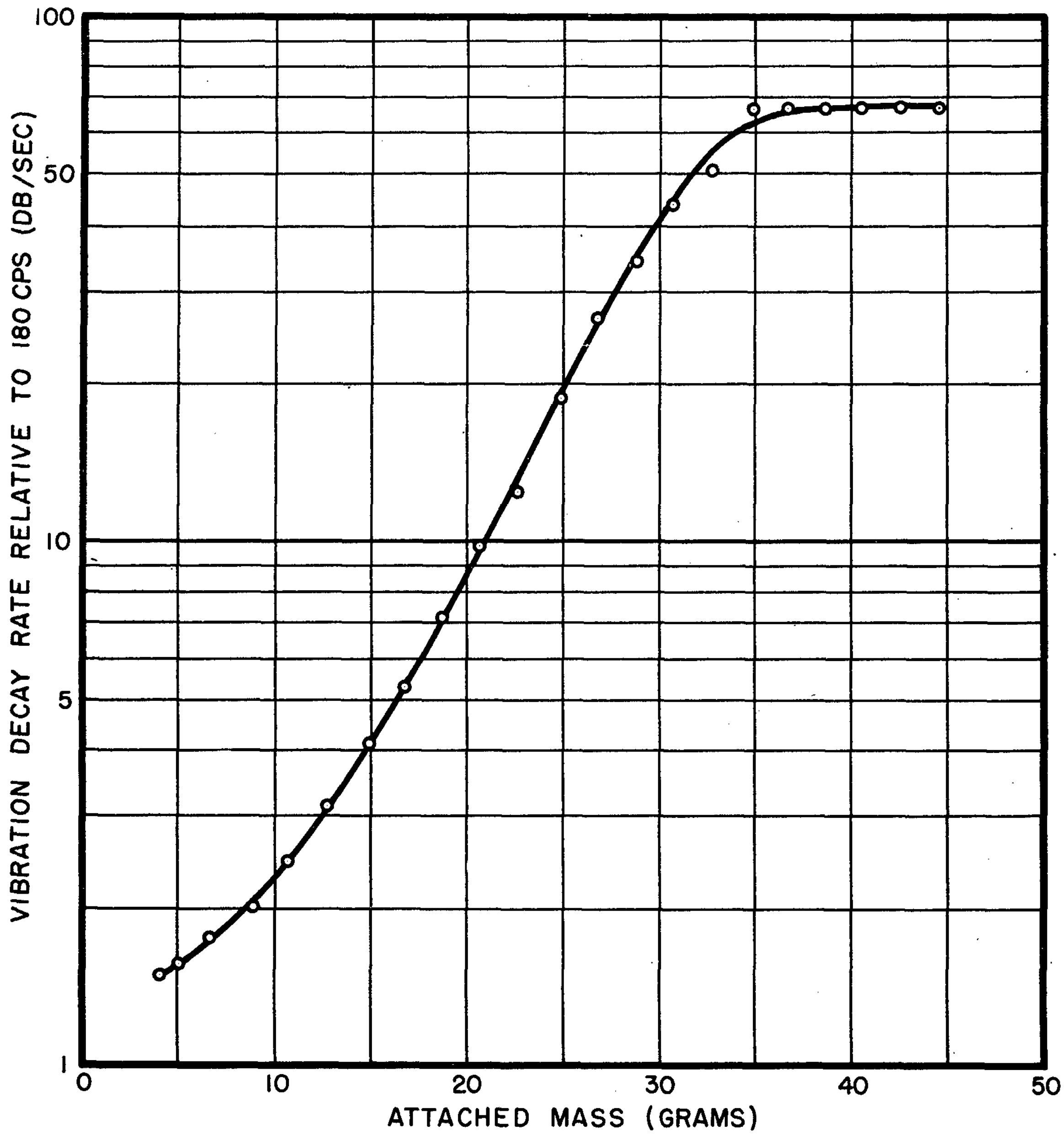

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
180 C.P.S.
100
50
10
5
1
0
10
20
30
40
50
VIBRATION DECAY RATE (DB/SEC)
ATTACHED MASS (GRAMS)

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
146 C.P.S.




3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
129 C.P.S.
100
50
10
5
1
0
10
20
30
40
50
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
ATTACHED MASS (GRAMS)

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
113 C.P.S.



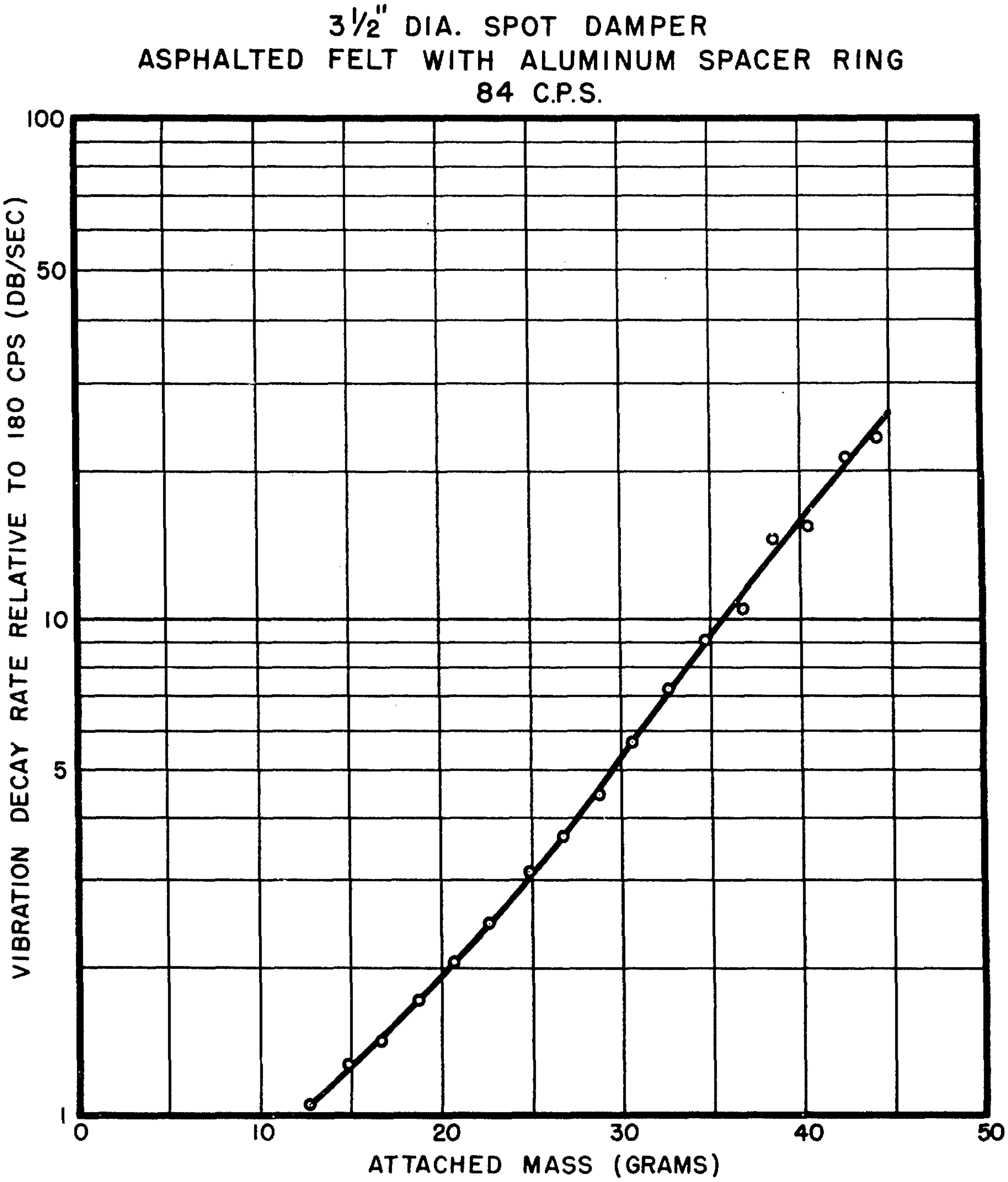

3½" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
84 C.P.S.
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

FIG. 27

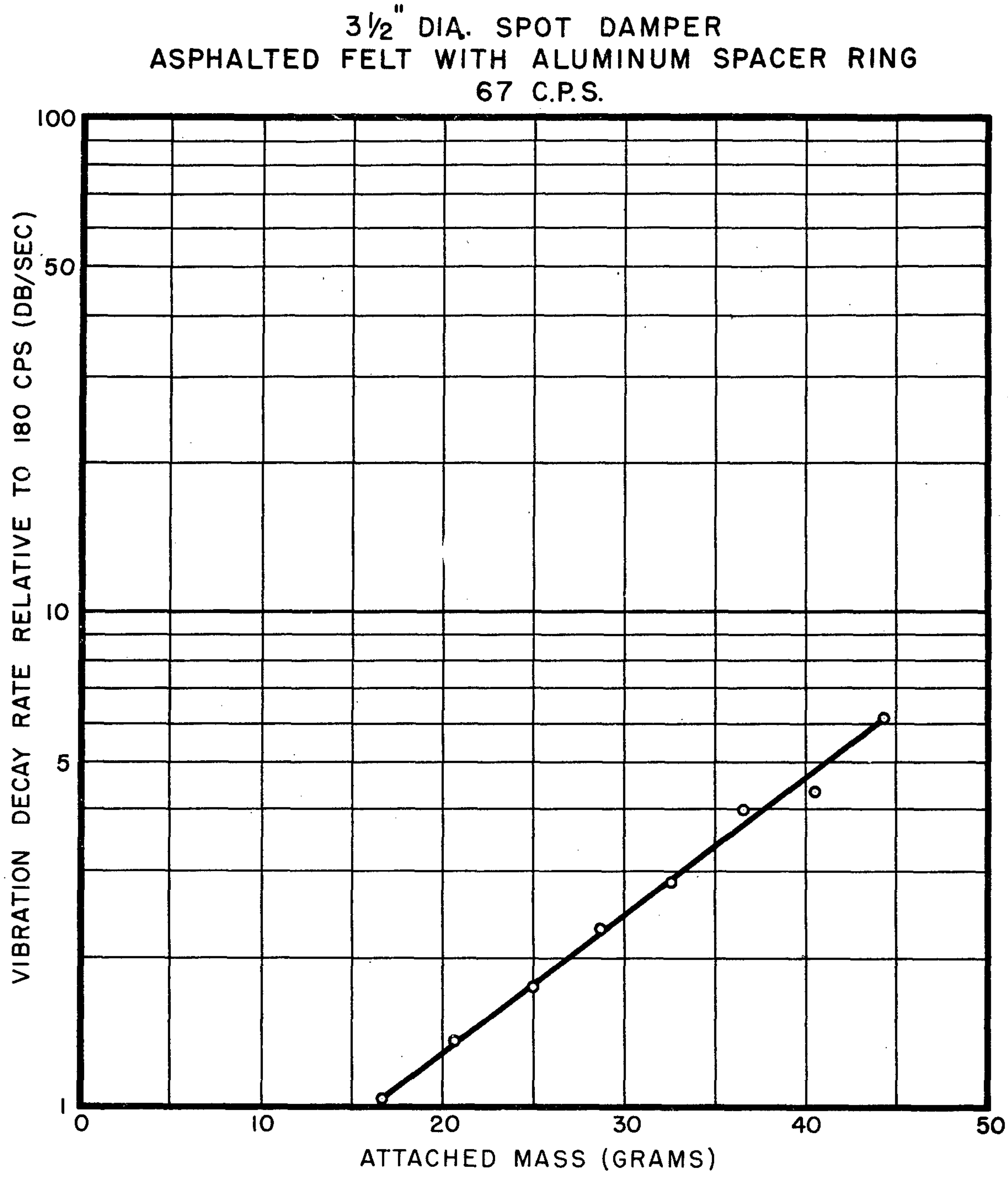
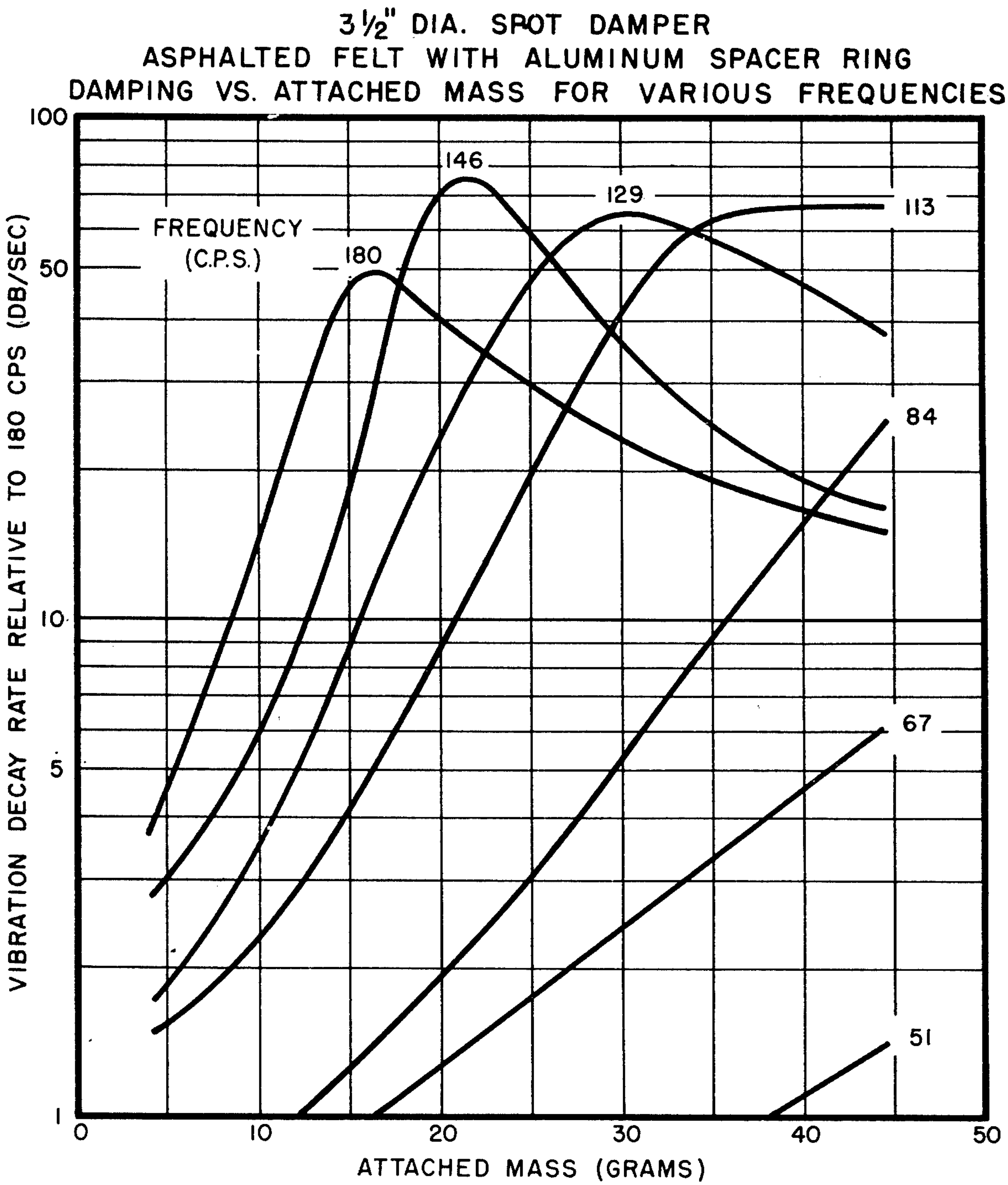
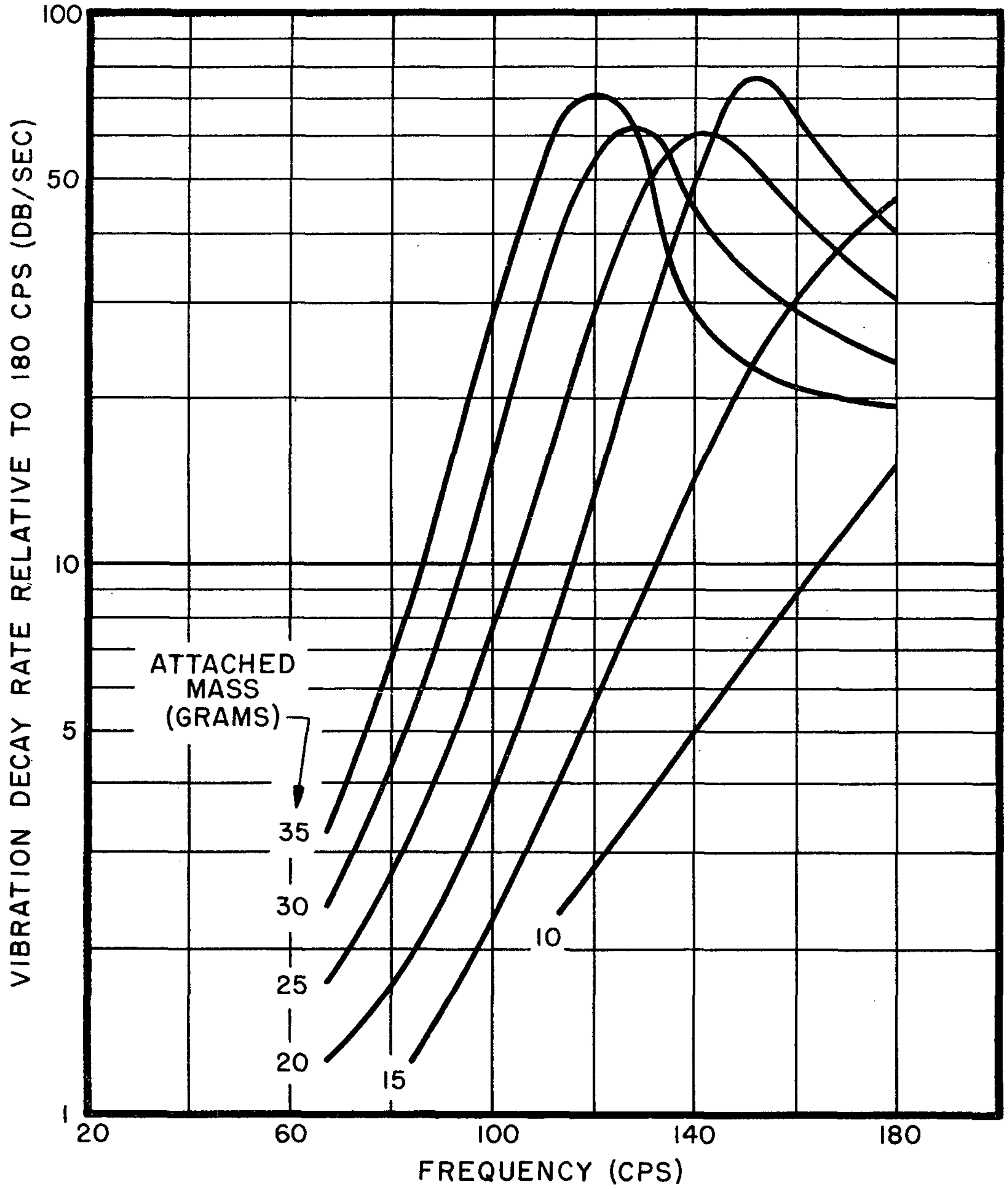

3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
67 C.P.S.
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
100
50
10
5
1
0
10
20
30
40
50
ATTACHED MASS (GRAMS)

FIG. 28

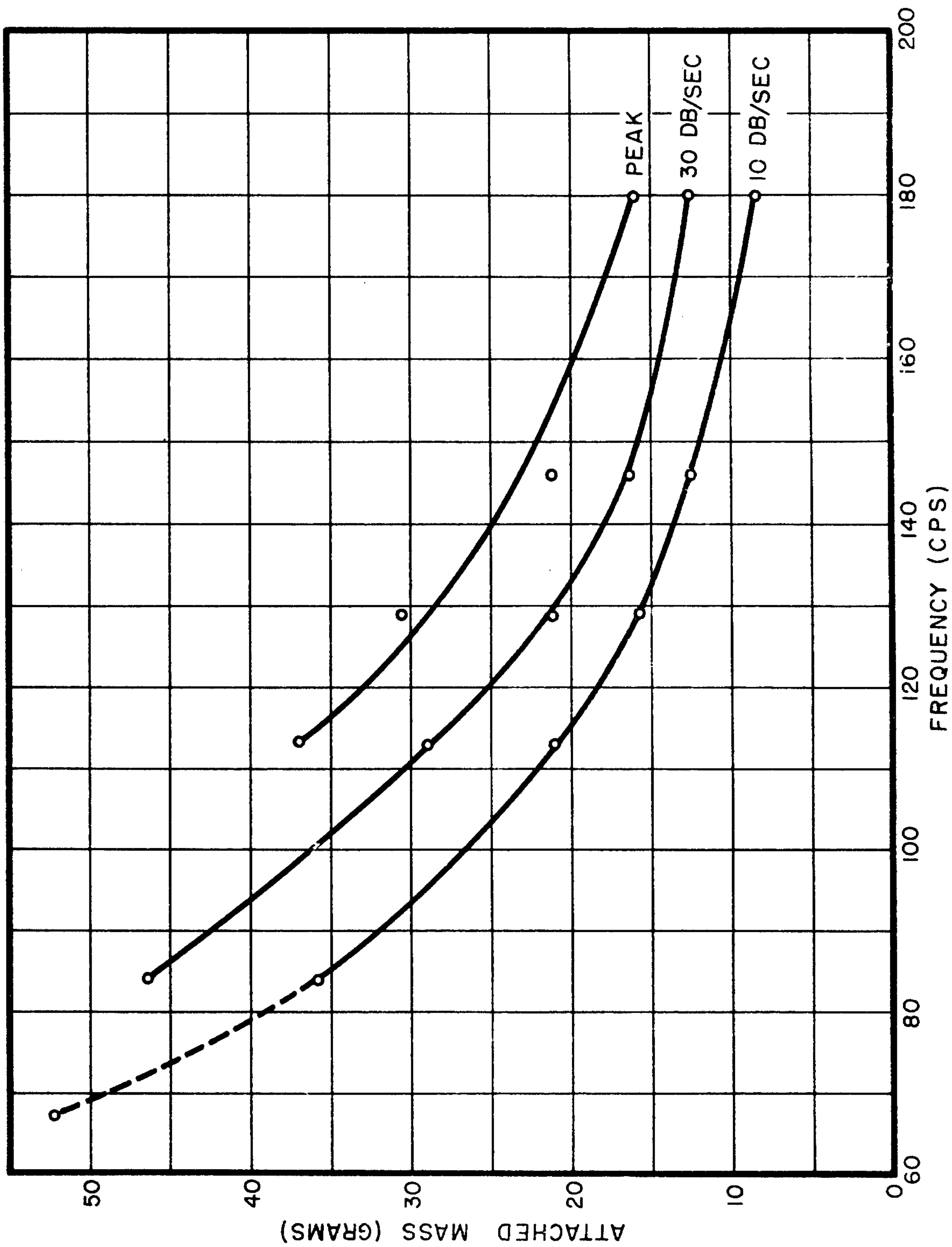


3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
DAMPING VS. ATTACHED MASS FOR VARIOUS FREQUENCIES
100
50
10
5
1
0
10
20
30
40
50
FREQUENCY
(C.P.S.)
180
146
129
113
84
67
51
VIBRATION DECAY RATE RELATIVE TO 180 CPS (DB/SEC)
ATTACHED MASS (GRAMS)

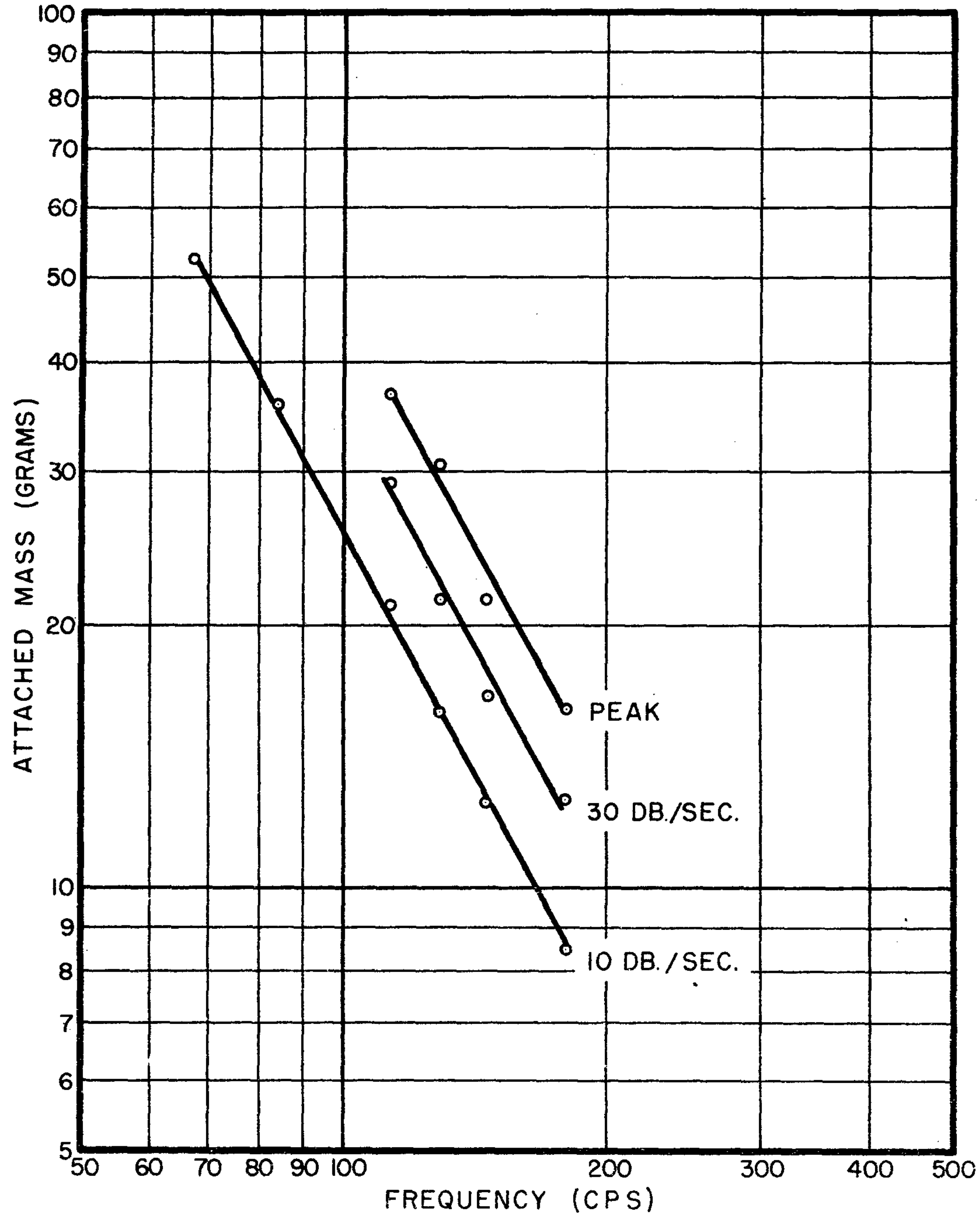
3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
DAMPING VS. FREQUENCY FOR VARIOUS ATTACHED MASSES



3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
ATTACHED MASS VS. FREQUENCY FOR VARIOUS AMOUNTS OF DAMPING



3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
DETERMINATION OF TUNING LAW.



3 1/2" DIA. SPOT DAMPER
ASPHALTED FELT WITH ALUMINUM SPACER RING
DAMPING VS. TEMPERATURE, 180 CPS, 26.72 gm.

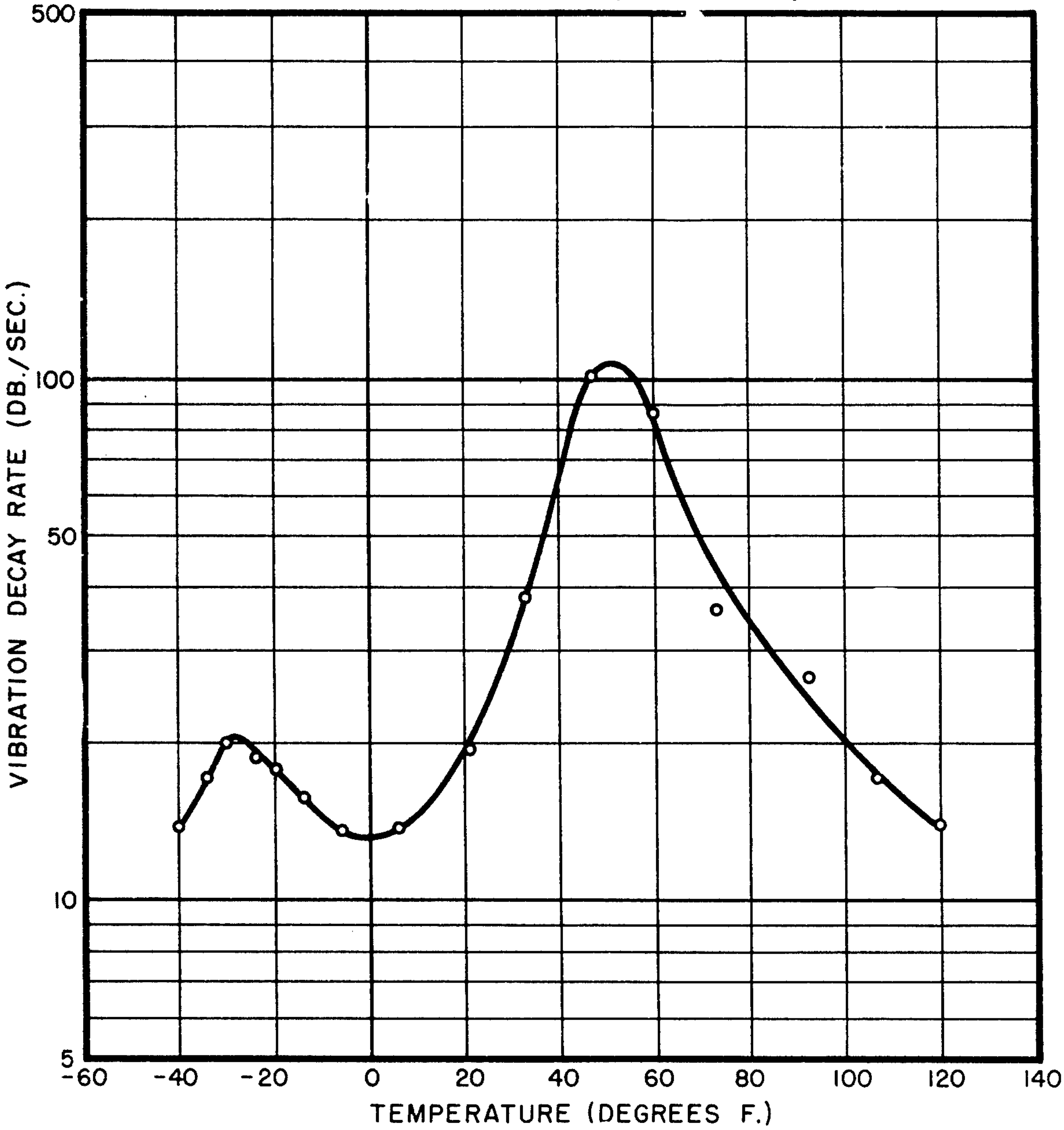
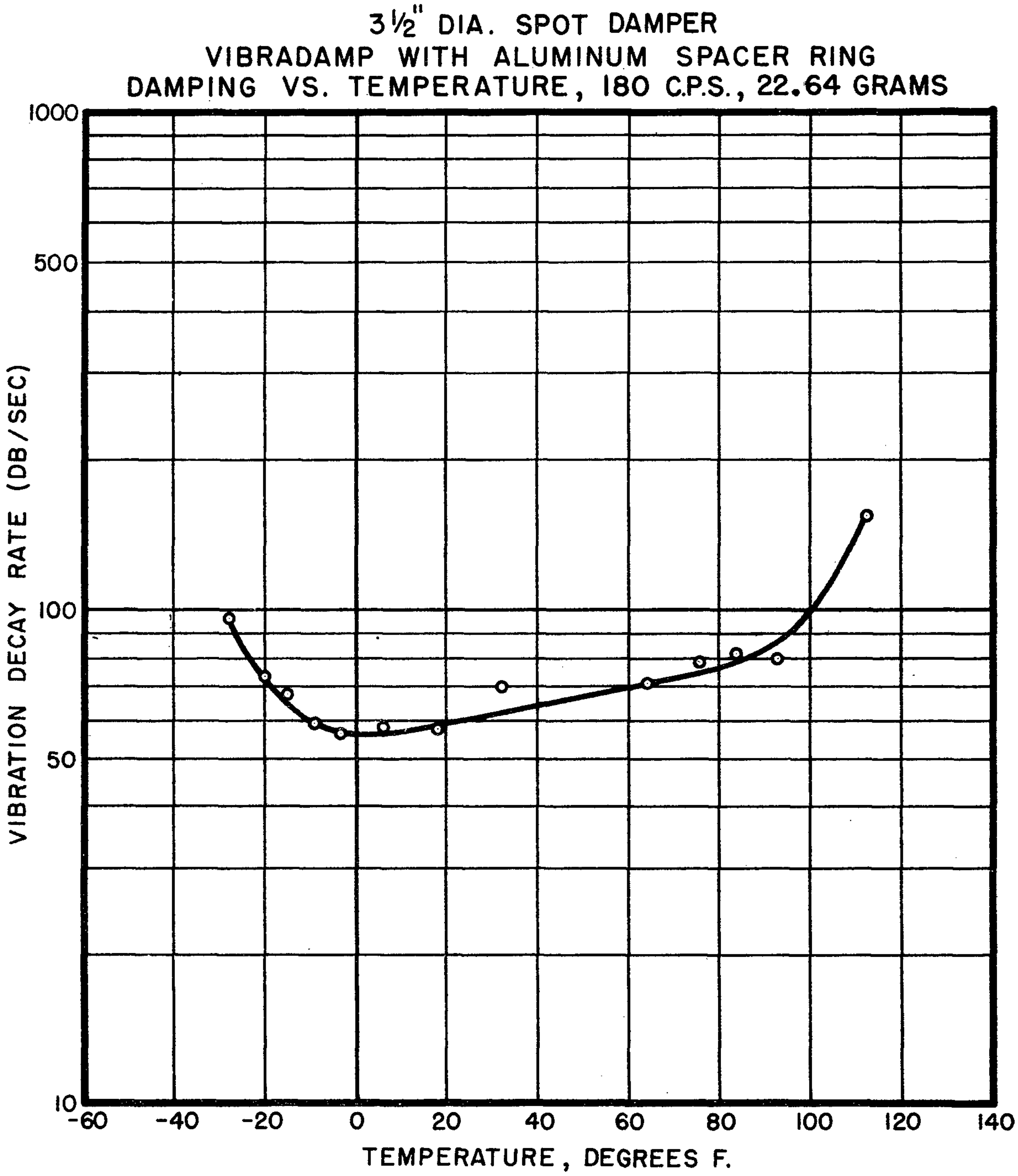


FIG. 33



3½" DIA. SPOT DAMPER
VIBRADAMP WITH ALUMINUM SPACER RING
DAMPING VS. TEMPERATURE, 180 C.P.S., 22.64 GRAMS
VIBRATION DECAY RATE (DB/SEC)
1000
500
100
50
10
-60
-40
-20
0
20
40
60
80
100
120
140
TEMPERATURE, DEGREES F.